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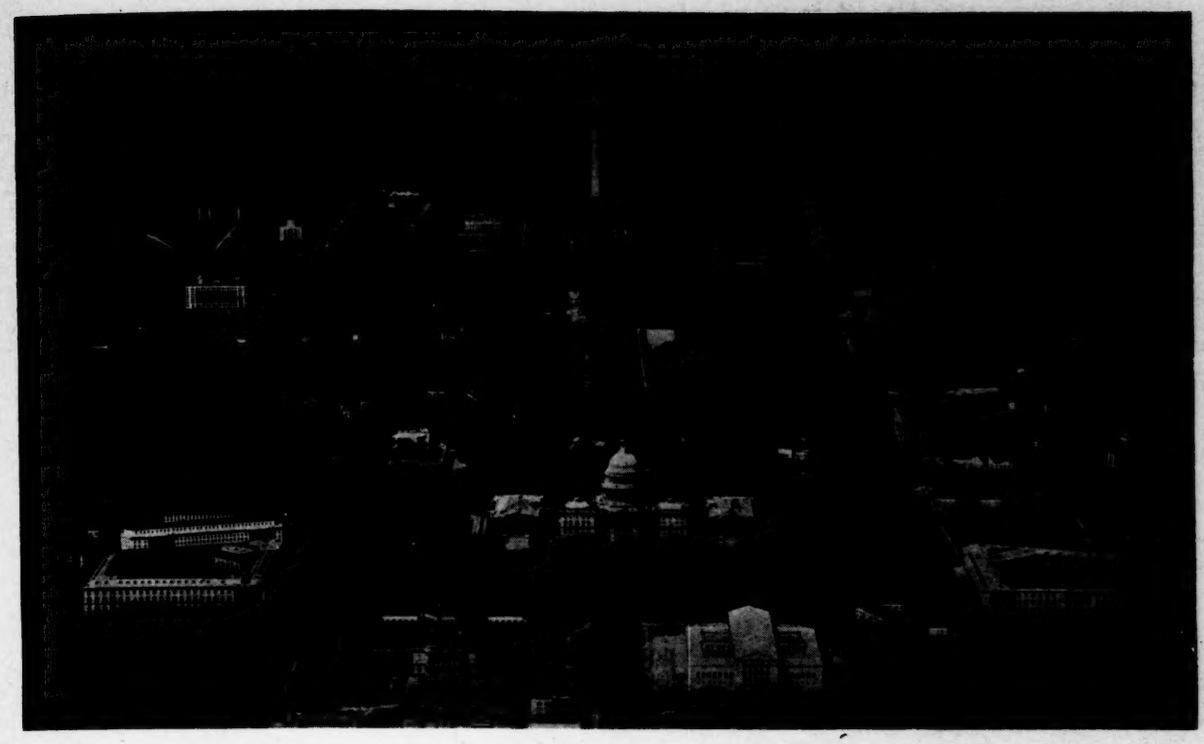


# Water Temperatures in Washington, D. C.

By MARY O. SOROKA

REGARDLESS of the type of air conditioning system adopted, the question of an adequate supply of cold water is of primary consideration and a discussion of water is not out of place in a study of weather when it is recalled that the temperature of the air determines the temperature of the water too.

According to U. S. Geological Survey "the temperature of the water in the ground at any place is in general about the same as the mean annual temperature of the air. Near the surface the temperature of the water follows the temperature changes of the air; at greater depths the water temperature corresponds to the increase in the earth temperature with increasing depth." In Washington the mean annual temperature is 55F for a 49-year period, but the lowest well water temperature on record is 58F.

Table on following page indicates the range in well water temperatures in the District of Columbia region. These values are as received from the owners or well drillers. No attempt was made to check them with a standard thermometer and no information about the time of the year at which they were taken was available.

Underground conditions are extremely variable from place to place and no general conclusions can be drawn. Borings indicate layers of alluvial sands, gravel, silt and clay which occur in lenses that vary radically in short distances. This material rests on a floor of ancient crystalline rocks. The best known water-bearing stratum in Washington is the zone which lies directly on the floor of the rock but even this zone cannot be located everywhere. An abundant supply of well water at one location does not necessarily mean that there is a similarly good supply across the street.

We depend upon the atmosphere for our entire water supply and it is fed from the clouds to the streams as rain and snow. Underground waters in the District are reservoirs of rain water. This water falls on an open area and seeps down through the soil until it finally collects in beds of coarse sand and gravel. There are two natural limitations to the continued success of even the best drilled well and these are first that the supply of water will be depleted after a short period of use because the infiltration is not sufficiently rapid after the existing accumulation of many years' rainfall has been exhausted, and second that a year of drought will cut off the infiltration completely.

In the downtown business district there is the possibility that drainage operations incidental to heavy construction will draw down the water level of wells nearby. In fact, the heavy pumping of well water for air conditioning itself may deplete the supply eventually. These are some of the reasons why the supply of water in a well is not necessarily permanent after it has been put in.

**The author shows that in one locality, at least, the temperature of surface water fluctuates very closely about the dry-bulb air temperature and in addition gives some information gained through practical experience with well water temperatures in a large city. The material in this article is based on data collected by the author and Robert B. P. Crawford, air conditioning engineer of Washington, D. C.**

| Location | Depth of Well, Ft. | Temperature, F |
|---|---|---|
| 14 and E Sts., NW. .. | 70 | 65-68 |
| 13 and F Sts., NW. .. | 70 | 58 |
| Relay, Md. ..... | 50 | 51 |
| Langdon, D. C. ..... | 210 | 60 |

It has been possible to secure quantities up to 300 g.p.m. from one well in the downtown section but this is not the usual condition. The ordinary expectation for a 10-in. well in water bearing material would be 150 to 200 g.p.m. An installation of two wells would not yield much more than a single well if they are close to each other.

An abundance of surface water is no indication of an abundance of underground water; it often means the opposite. Before deciding to use the surface water in a shallow well, an investigation of possible foundation difficulties resulting from lowering the water table should be made. This might cause enough shifting of soils to produce settlement. It is not the draining off of clear water that is harmful but pumping sand along with it thereby producing voids or dissolving out some clay which might be in the sand and gravel beds. These are the causes for the earth movements which crack buildings.

However, no investigation of supposed geological conditions nor previous local experience, nor even the opinion of well drillers can compare with the results secured from the core boring in a test well. Little water is to be found in the rock but there is the possibility of striking a crevice in the rock which often yields a good supply of water.

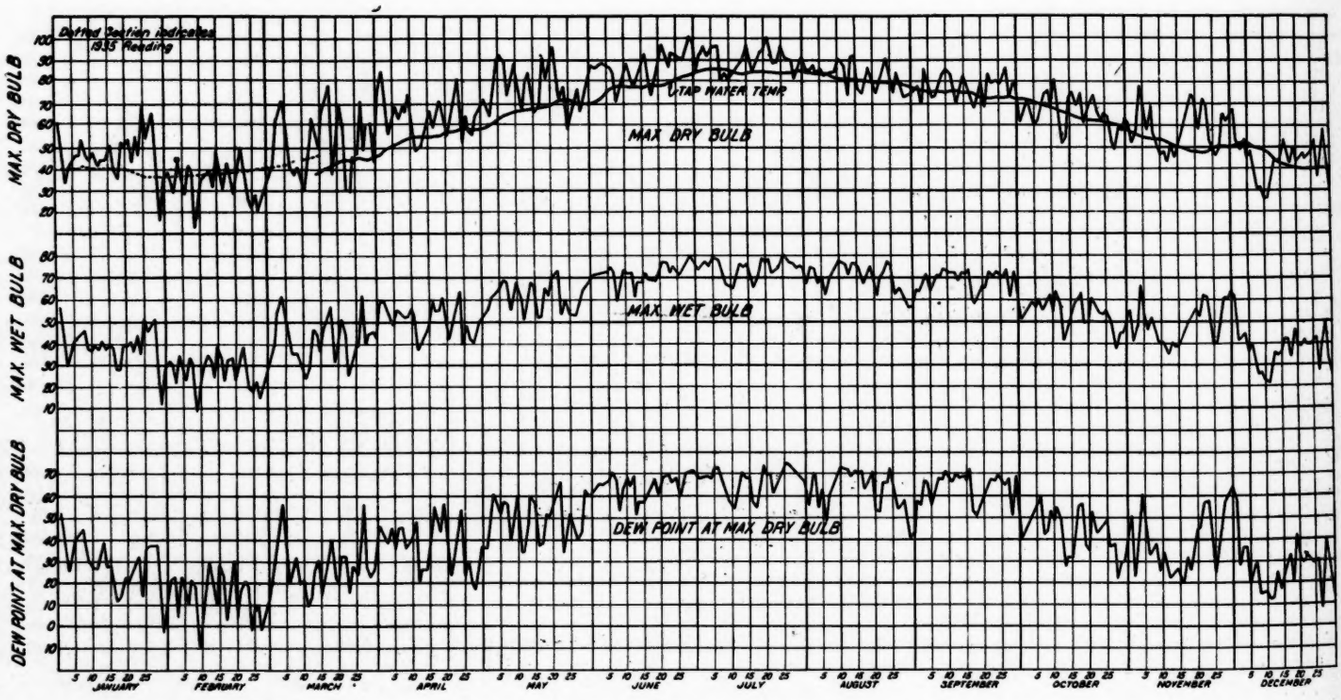
The chief factor in the success of the well is the screen, next in importance is the method of development and finally the position of the pump with respect to the screen. Servicing of wells is usually not necessary in wells which are properly installed. The technique of well drilling has advanced to such an extent that it is possible to install a well with as little headroom as the height of the basement story.

Aside from the saving when purchasing water, another advantage of the use of well water is the economy secured by splitting up the load into three different functions, each operating at its most efficient temperature. By this method, the air can be precooled and dehumidified by the 64F water before it is used for condenser purposes. Of course a plant must be large enough to warrant staging of the load. For example, air coming in at a temperature of 80F wet bulb and reduced to 52F wet bulb requires 21.4 B.t.u. per lb. of air. When air is received at a temperature of 65F and reduced to 52F wet bulb, the requirement is only 8.4 B.t.u. per lb. which means a saving of 13 B.t.u. for each pound of air used.

### City Water Supply

Aside from the cost, use of the city water supply of Washington is somewhat limited because the temperature of the water in the mains reaches 86F in the summer. It is often said that water is conservative in the matter of temperature changes, but a year's record of tap water readings, shown in Fig. 1, indicates changes in temperature of the Potomac River corresponding to the change in the daily dry bulb temperature. In 1933-1934, tap water had a range of 50F. The maximum reached was 86F on July 7 and the minimum was 36F on Feb. 1. The Tidal Basin, which is the source of supply of some of the Government buildings, also reaches 91F in the summer but cools off about 5F in the pipe lines.

For the water temperatures shown, if a well is used, 1.5 g.p.m. are required per ton or per horsepower, for city water 3 g.p.m. and for cooling tower 4 g.p.m. per ton or per horsepower.




*Records for Washington for the year 1934 showing the comparison of city water temperatures with mean dry bulb temperatures*

# Heating and Ventilating Requirements for Housing

By C.-E. A. WINSLOW†

**The subject of housing is much in the public eye. With the U. S. Housing Authority actively attacking the problem of low-cost housing and with the government's loan insurance plan for new private construction liberalized, we may expect more and more activity in both public and private housing. For these reasons, Doctor Winslow's excellent summary of the standards and methods in heating and ventilating for housing is particularly timely. The paper is abstracted from one which the author presented before the Symposium on Hygiene of Housing at the recent annual meeting of the American Public Health Association in New York.**

WE cannot intelligently consider either heating or ventilation without knowing what sort of space we are to heat or ventilate. Therefore, a basic factor in the hygiene of housing is the establishment of a minimum space allotment.

The ordinarily accepted standards for decent living (as stated in 1934 by the National Association of Housing Officials) specify 500 cu. ft. of air space for each adult and 250 cu. ft. for each child in any occupied room. We may perhaps take 400 cu. ft. as a rough mean value. This standard, however, refers to occupancy at any given moment and the home should provide separate accommodations for day and night occupancy. A total of 800 cu. ft. per person, should, therefore, be considered a minimum for the dwelling as a whole.

This figure is somewhat below the standard recommended by the Housing Commission of the League of Nations at its meeting last June which states that on hygienic grounds a living room used in the day-time or a sleeping room at night should each provide 525 cu. ft. per person and, if sleeping and living rooms are separate, a total of at least 875 cu. ft. per person for the total dwelling is essential. The Polish delegation presented, at this meeting, very interesting evidence in support of these figures, based on actual estimates of space required for necessary furniture and equipment.

For a family of five persons we shall require, then, at least a combined living room and kitchen having a minimum capacity of 2000 cu. ft. At least two bedrooms must be provided with the same total capacity. We may consider this home for convenience as a building (or section of a building) 20 ft. wide by 25 ft. long and 8 ft. high. This corresponds to the minimum standards of the English law which define a dwelling as overcrowded if it has less than three separate rooms for five persons.

Such a minimum dwelling would then include one room with a floor space of 250 sq. ft. and two bedrooms, each with a floor space of 125 sq. ft. Tenement regulations in such cities as New York call for one room in each dwelling having a floor space of 120 sq. ft. and establish a minimum of 70 sq. ft. for all other rooms. If individual rooms were made as small as this, the number of rooms could be increased, but the total 4000 cu. ft. for a family of five seems essential for health and decency. These are overall inside dimensions and include toilet and closet space.

### Heating Standards

The maintenance of an adequate temperature during the cold season is clearly an essential factor in the hygiene of housing. The thermal influence of the atmosphere upon human health and comfort depends upon four distinct factors; the dry bulb temperature of the air, the humidity of the air, the degree of air movement, and the mean radiant temperature of the surroundings (including walls, flooring, ceiling or roof, windows, heating appliances, and human bodies). Various attempts have been made to combine two or more of these factors in a single index of human well-being such as the ASHVE "effective temperature," the French "resultant temperature," and the English "equivalent temperature"; and various instruments, such as the katathermometer, eupatheoscope, globe thermometer, resultant thermometer, and thermo-integrator, have been devised to estimate such indices. None of these procedures has proved wholly satisfactory and the Housing Commission of the League urges that in research on this problem direct measurements of the four basic physical factors involved should be made wherever possible.

In the ordinary dwelling, however, air movement does not vary greatly and, if temperatures are within a comfortable range, relative humidity may vary between 25 and 70% without marked physiological effects. Hence, from a practical standpoint, air temperature and radiant temperature are the chief variables to be considered.

The ASHVE specifies as standard of ideal winter conditions an "effective temperature" of 68F which corresponds (with varying degrees of relative humidity between 30 and 70%) to dry-bulb temperatures of 68F to 72F. The ASHVE *Guide* points out that with cold walls the air temperature may have to be 2F to 6F higher.

These figures no doubt correspond to common American habits but it seems probable that they are considerably above real hygienic essentials. The range approved by English authorities is from 60F to 68F, and too sharp a contrast between indoor and outdoor temperatures may well play a part in the high incidence of respiratory disease in the United States.

†Professor of Public Health, Yale School of Medicine, New Haven, Conn.

It would be highly desirable to base our judgments on the combination air and wall temperatures which obtain in a given case, and the John B. Pierce Laboratory of Hygiene has recently introduced a measure designated as "operative temperature"[1] which measures this combined effect on a purely physical basis and without assumptions as to comfort reactions. Standards are likely, however, for some time to be based on ordinary dry bulb temperatures. So long as this is the case, the dwelling should be provided with equipment which will maintain an air temperature of 68F-70F (or of 73F-75F, with two or more exposed walls) during periods of occupancy of a given room.

It is by no means necessary or desirable, however, that these temperatures should be maintained in sleeping rooms at night and temporary radiant heating for such rooms should be given serious consideration.

### Insulation

The problem of heating (in a given climate) is conditioned on the one hand by the interior temperature to be maintained and on the other by the degree of heat loss through walls, windows, ceiling and floor. It need only be pointed out here that weatherstripping of window frames and doors, the installation of double windows, and the use of insulating materials in walls and ceiling may—under cold climatic conditions—reduce fuel costs by more than half. Whether such insulating procedures will save more than they cost depends on climate, on local fuel and construction costs, and on the degree to which radiant heating can be utilized.

[1]C.-E. A. Winslow, L. P. Herrington, and A. P. Gagge, "Physiological Reactions of the Human Body to Varying Environmental Temperatures," Amer. Jour. Physiol., 1937.

### Heating Methods

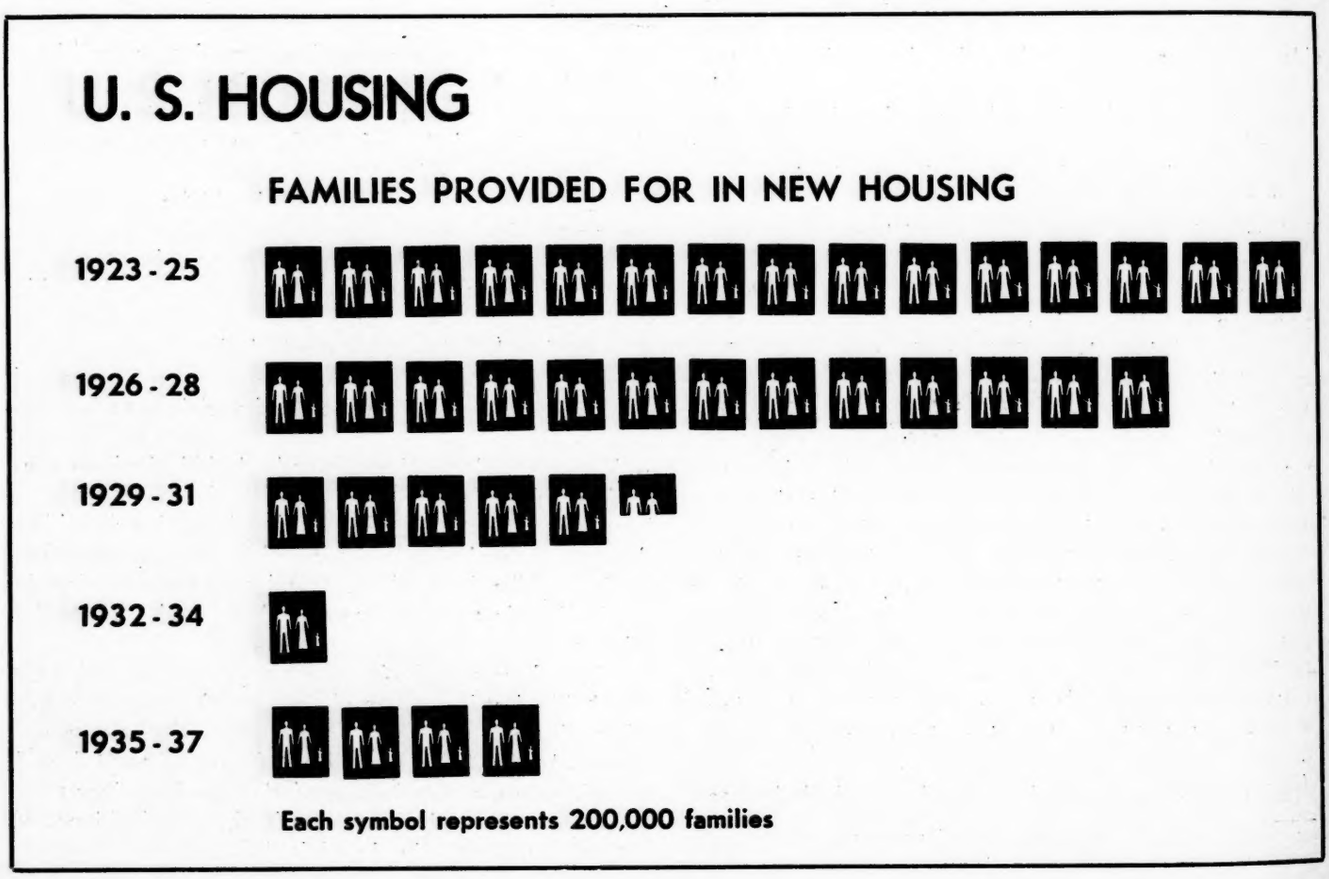
The problem of specific heating appliances for the low-cost dwelling must, in general, be solved differently in the isolated house and the block of tenements, respectively.

In the case of the isolated house the cost of construction for low-income groups would prohibit the inclusion of a basement, except perhaps a small earth-floor cellar for the winter storage of vegetables and the like. The absence of a basement would limit heating possibilities to fireplaces, stoves, and small hot water circulating systems.

Fireplaces would be unduly expensive from the standpoint of construction and even more wasteful in operation, except in regions where the winter climate is exceedingly mild.

A circulating heater type of stove which can be obtained equipped to burn either wood, coal, oil, or gas would be the most desirable method of heating on a basis of first cost. A small house with sleeping quarters on the second floor can be heated by placing registers in the ceiling so that the warm air from the lower floor will rise to the second floor.

A small hot water heating system with hot water radiators could be substituted for stoves, but at greater expense. With apartments, and sometimes perhaps with blocks of one- or two-family houses, steam heating comes into the picture as an economic possibility. The saving of fuel with such a system (as compared with stove heating) will amount to at least one-fifth, and, with anthracite coal, to half the total cost of heating—a saving which goes far toward paying interest on the additional capital investment involved. The general



evenness of heat distribution and the freedom from fire hazards make steam heating the method of choice for multiple-family dwellings.

It is impossible at present to specify the particular type of radiator which is most economical and most hygienic; but a design which gives off a considerable proportion of its heat by radiation is desirable since it produces a maximum effect within the zone of occupancy, and a design which minimizes vertical differences in temperature is also helpful as decreasing tendency to heat losses through the ceiling. Supplementary radiant heating may have great usefulness, particularly in rooms such as bedrooms and bathrooms where quick temporary heating is desirable but where high temperatures need not be consistently maintained.

### Ventilation

The standards of the ASHVE specify a supply of 10 c.f.m. of fresh outdoor air as a minimum, and this standard has been amply justified by the studies of Yaglou at Harvard, on the air change necessary to remove body odors. In an occupied space, such as we have suggested, which provides a cubic capacity of 400 cu. ft. per occupant it is obvious that only one and a half air changes per hour would be necessary to fulfill the minimum requirement. This degree of air change would ordinarily obtain as a result of infiltration, in a room with two or three exposed walls in cold or windy weather and could easily be attained under any conditions by intelligent use of windows. No provision for forced ventilation is therefore necessary in the ordinary low-cost home, providing

A. That the cubic air space specified above is available for both sleeping rooms and living rooms;

B. That windows are adequate and properly placed;

C. That reasonable regulations are in force with regard to distance between buildings so as to permit of free circulation of air in summer.

It does, however, seem essential that the range used for cooking should be provided with a hood and special exhaust duct to the outside air in order to eliminate odors and excess heat. Gas heaters or other heating appliances, if not connected with a flue, should be provided with exhausts. The toilet room need not be specially ventilated if it is provided with a window opening to the outside, which we consider to be essential on general grounds of cleanliness.

The cooling of occupied spaces during the warm season has material hygienic significance, but does not as yet appear to be practical for the low-cost home. The inclusion of one electric fan in the equipment of such a home would, however, be fully justified on hygienic grounds.

### Summary

The essential hygienic features of low-cost housing construction from the standpoint of heating and ventilation may be summarized as follows:

1. Provision of a cubic capacity of 500 cu. ft. of space for each adult and 250 cu. ft. for each child in every occupied room. Where living rooms and sleeping rooms are occupied at different times, the figure per room must be doubled for the entire house or apartment, giving a mean total allowance of 800 cu. ft. per person.

2. Construction, which provides such a degree of insulation (by weatherstripping and double-glazing of windows, insulation of walls and ceiling) as is economically warranted by local climatic conditions and fuel costs.

3. The installation of heating appliances (circulating heater stoves in the isolated dwelling and one-pipe steam heating in blocks of tenements) adequate to maintain a dry bulb temperature of 68F-70F (or an equivalent operative temperature) in rooms for day occupancy. Sleeping rooms can be advantageously heated by radiation for temporary purposes.

4. Local ventilating flues to remove heat and fumes from the range and from any heating sources which involve danger of carbon monoxide production.

5. Provision of at least one electric fan for cooling where summer heat is considerable.

---

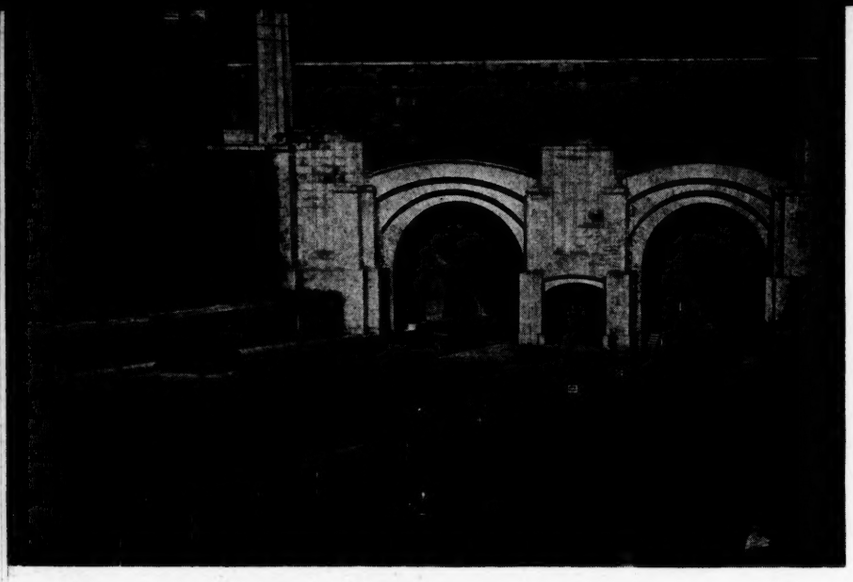
## Effect of Barn Temperatures on Milk Production

WHILE it is known that temperatures maintained in farm buildings have an effect on the production of milk, not much work has been done to find the conditions most suitable from the milk production standpoint. Some indication of the temperatures which should be maintained and of effect of cold weather on milk production was included in a report on the relation of stable environment to milk production prepared by M. A. R. Kelley, Bureau of Agricultural Engineering, and I. W. Rupel, University of Wisconsin.

The report was based on studies and tests made on 88 cows housed in four Wisconsin stables during the winters of 1930-31 and 1931-32. The cows in the tests were selected from a herd of 600 to form groups balanced as nearly as possible as to age, size, productivity, stage of lactation, and other factors.

Under Wisconsin winter conditions the optimum stable temperature for cows in stanchions and on moderate feeds appeared to be about 50F. There was little decrease in production when the stable temperature was 45F. Production of milk was above the average when the temperatures were 60F to 65F but the butterfat was below average. At warmer temperatures the cows seemed less comfortable than at lower temperatures.

The report concluded that there is no economic justification or need for maintaining stable temperatures of 60F to 65F if excessive insulation or artificial heat is required to do it. A uniform temperature of 50F to 55F was found satisfactory, the lower stage being conducive to good appetite, body vigor, and a healthful stable condition.



*Weehawken, N. J., entrance to the tunnel. This tunnel is similar to the Holland Tunnel, the principal difference in the ventilating systems being that the Lincoln has three ventilation buildings instead of four for the Holland. The Lincoln has a vitrified brick roadway surfacing and glass tile ceiling.*

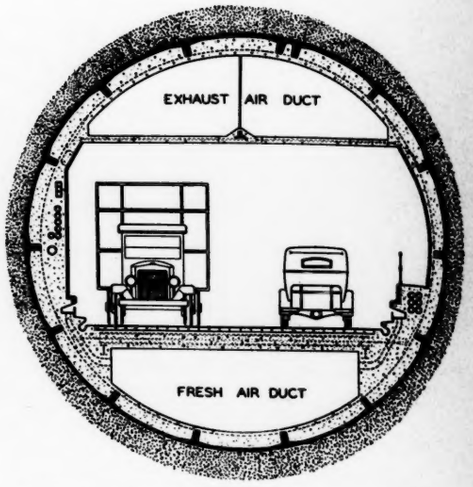
THE Lincoln Tunnel, connecting 39th Street, Manhattan, with Weehawken, N. J., is a third interstate motor highway across the Hudson River in the New York City district. It was constructed by The Port of New York Authority to serve the area midway between the Holland Tunnel and the George Washington Bridge.

A major engineering feat, the Lincoln Tunnel opened to traffic December 22, 1937, three years and seven months after the start of work. Two shields (steel cylinders with sharp cutting edges propelled forward by hydraulic jacks) bored through the river bed, starting from opposite sides and met within one quarter of an inch when the tunnel was "holed through."

Photos of entrance, motors and switchboard by Galloway; others by The Port of New York Authority.

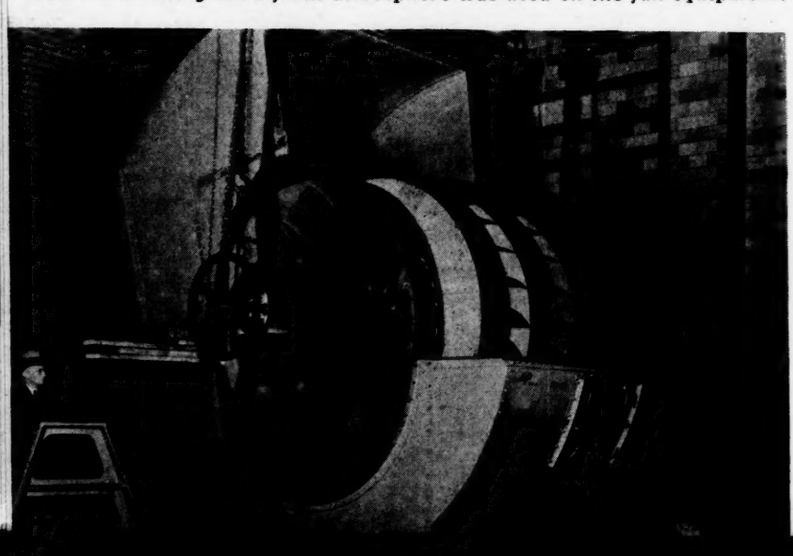
# VENTILATION OF THE

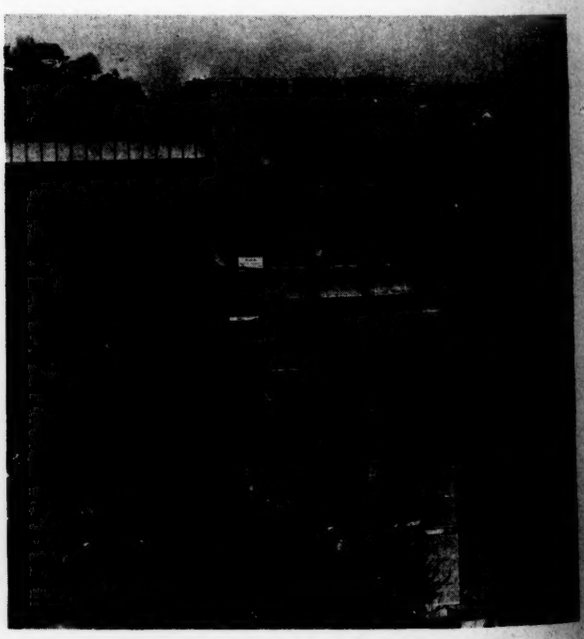
*One of the direct-connected motors for the river section ducts. A large two-speed motor is coupled by means of a flexible coupling to the fan shaft, and a smaller, geared, two-speed motor connected to the extended shaft of the larger motor through a flexible overrunning clutch. When the larger motor is operating the fan, the geared motor is out of service. For lower speeds the geared motor drives the large motor and fan. With this, low speeds are possible with high efficiencies.*




*Sketch showing location of supply and exhaust ducts relative to the roadway. Air is introduced into the roadway area through an expansion chamber and a narrow adjustable slot located above each curb and spaced from 10 to 15 ft. apart throughout the tunnel.*

*Setting a supply fan in the New Jersey ventilation building. Records of the Holland Tunnel show that the fans are operated at approximately half speed for 80% of the time. Squirrel cage motors were specified. A special aluminum paint which will resist the corrosive effect of New York's salty waterfront atmosphere was used on the fan equipment.*



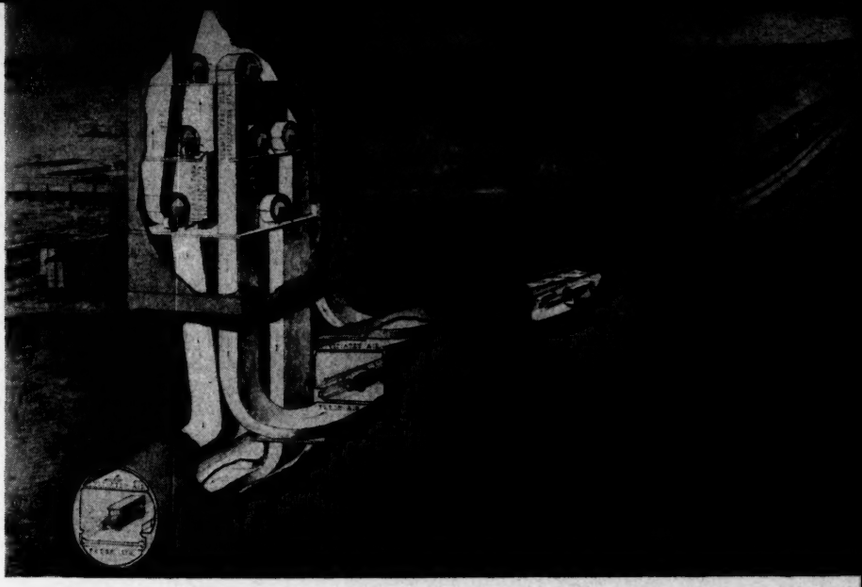


*The ventilation building at the Weehawken, N. J., side looking northeast. Louvers are built into the north, south, and east walls and extend from about the second floor to the fifth floor. Louvers are of the motor-operated, rolling-shutter type.*

## DATA ON THE LINCOLN TUNNEL

| | |
|---|---|
| Length of first (south) tube, portal to portal | 8,215 ft. |
| Length of each tube under river | 4,600 ft. |
| Width of two-lane roadway, between curbs | 21½ ft. |
| Operating headroom clearance | 13 ft. |
| External diameter of iron and steel shell | 31 ft. |
| First unit opened for traffic | Dec. 1937 |
| Ultimate cost of complete project | $74,800,000 |
| Maximum depth from river surface to top of tunnel | 75 ft. |
| Three ventilation buildings with a total of | 32 fans |
| Number of supply fans | 26 |
| Number of exhaust fans | 30 |
| Air supply, south tube, c.f.m. | 1,736,000 |
| Air supply, north tube, c.f.m. | 1,650,000 |
| Operating hp., fans, first unit | 2,237 |

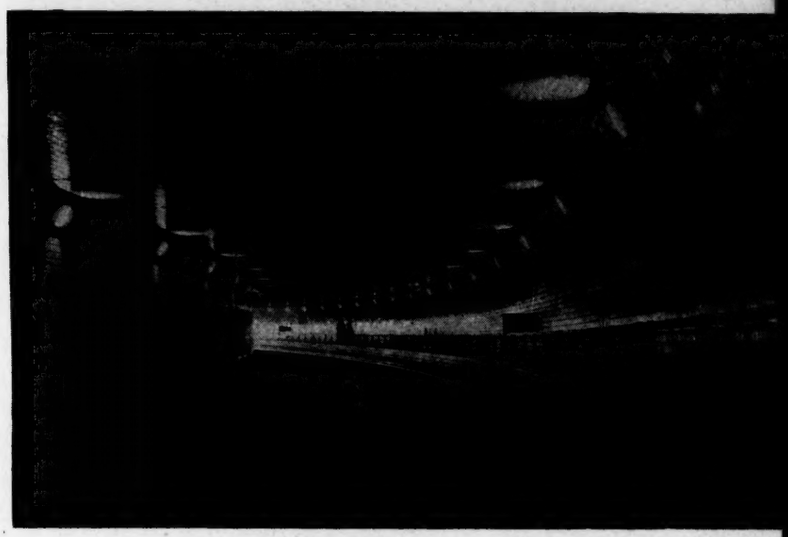
# NEW LINCOLN TUNNEL



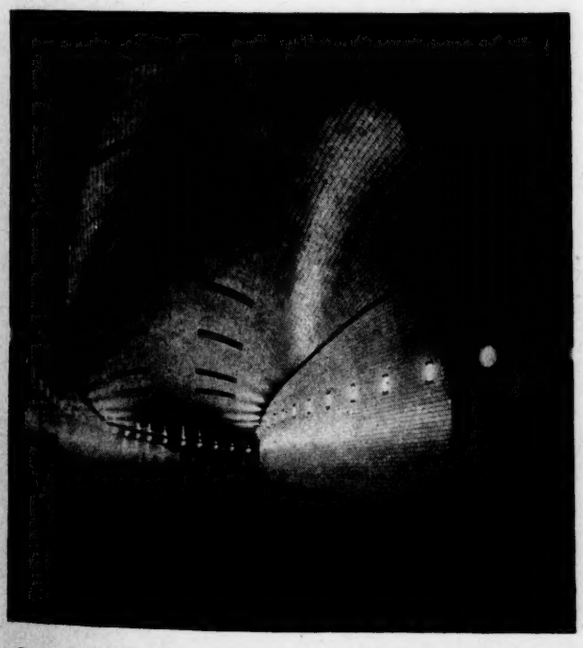
*Sketch of the ventilating system of the tunnel. Dampers are provided in the ducts extending between each fan and the common duct leading to the tunnel from the ventilation building. These dampers can be raised or lowered by motors.*

*Interior of the south tube of the tunnel looking west from a point west of the New Jersey ventilating shaft. Maximum rates of fresh air in the tunnel range from 150 to 275 c.f.m. per lineal foot of tunnel depending on the grade of the roadway. The maximum is provided on upgrades, the minimum on downgrades.*

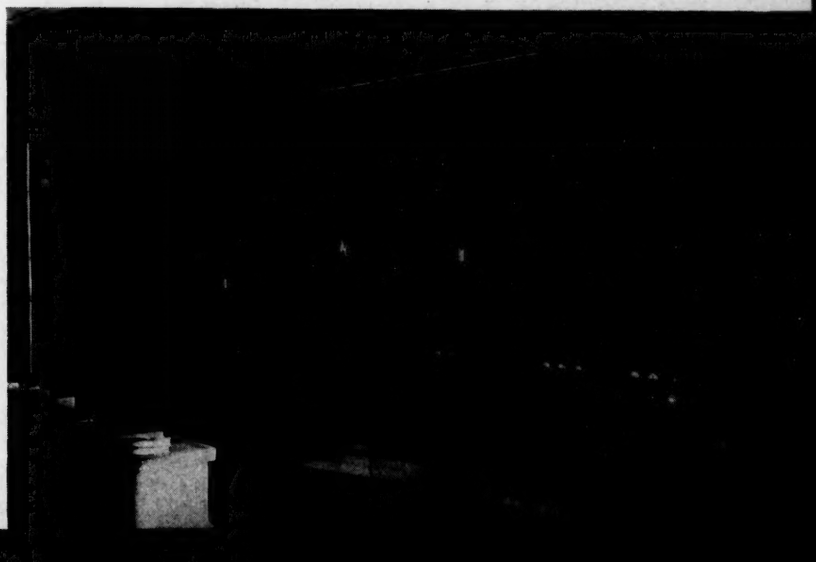
*Sketch of the arrangement in a ventilation building. The tunnel is divided into five sections for ventilation purposes, each being served by one fresh air duct and one exhaust air duct. By this method, longitudinal air currents, which would be a hazard in spreading smoke and fumes, are eliminated.*



*The central control board in the supervisory control room of the Weehawken ventilation building where the controls for ventilation, heating, lighting signals, and pumps are located. Parts of the ventilation building are heated. The toll booths are heated by unit heaters in order to keep the legs and feet of the toll collectors warm since the doors and windows are open most of the time.*



*Looking into the south tube from the New Jersey portal. This tube will be used for two-way traffic until the north tube is completed. Measuring stations are provided through the tunnel at every three hundred feet. These measure the air velocity and pressures*



By L. W. CHILD
*Evans Products Company*

# Summer and Winter Air Conditioning of Automobiles and Buses

**As the author points out, the air conditioning of an auto or bus is a difficult problem. The space per occupant is extremely small; infiltration may be very high at moderate velocities, and the allowable space for equipment is quite limited. Mr. Child outlines the problem, shows how it has been attacked up to the present time, and makes some predictions regarding future developments. This article is abstracted from a paper presented at the annual meeting of the Society of Automotive Engineers in Detroit January 10-14, and is reprinted here through the courtesy of the Society. Photo by Galloway.**

APPLICATION of air conditioning principles to an automobile or bus calls for somewhat different considerations than the problem of conditioning a home or public building. The space per person, especially in the bus, is considerably smaller than that found in the most crowded of buildings. The air for conditioning must be delivered in close proximity to the passengers, giving little chance for mixing it with the surrounding air before coming in contact with the occupants. Unlike the building, the car is moving at widely varying speeds causing correspondingly widely varying amounts of fresh air to leak into the car and so upset any carefully figured conditions. There is also the serious problem of available space for equipment.

Consider the relatively crowded condition in a five-passenger car as compared to a crowded restaurant. A restaurant 60 x 40 x 10 ft. high might, by dint of extreme squeezing, seat 200 persons. Each individual would then have 120 cu. ft. of space all to himself. Taking the passenger space in a five-passenger car as 150 cu. ft. the space per passenger is only 30 cu. ft. To equal this condition the restaurant would have to pack in 1000 patrons, which under its original table plan, would call for stacking them into four layers one on top of the other.

The leakage of outside air into a car or bus would seem to be an uncontrollable factor. It varies with the speed, the wind velocity and direction, the type of construction, and the age of the vehicle. The most logical way to attempt to control leakage is to supply enough outside air, properly conditioned, to cause the air in the body to leak out rather than in. Although this method appears to be impractical on very leaky cars, it is satisfactory on those produced within the last few years, and a distinct improvement in any vehicle, even though insufficient air is delivered to stop all inward leaks. Rough tests have shown that 60 to 80 c.f.m. of air delivered from outside into the body will stop leaks on most of today's four-door cars at speeds up to 60 m.p.h.

Considering the atmosphere inside the car, the most annoying phenomenon of winter driving is the formation of fog on the windows and the odor of wet woolen clothes and stale tobacco smoke. If we supply sufficient fresh air to eliminate the possibility of fog, experience has shown we will also eliminate the odor and smoke, stale or otherwise. Fogging of windows is caused by increased amounts of water vapor in the air due to that given off by the passengers. This can only be eliminated by keeping the quantity of water vapor in the air low enough to prevent condensation at the temperature of the glass.

A normal adult can be expected to give off 0.075 lb. of water vapor per hour, or 8.75 grains per minute, at temperatures below 70F. Although this figure is bound to vary, it has served satisfactorily in practice.

At speeds up to 60 m.p.h. rough tests show the minimum temperature of the inside surface of the laminated glass in a car to be two-tenths of the way between the outside and inside temperatures. For example, with an inside temperature of 70F and 0F outside the glass will be at 14F. This will vary with wind direction, speed, car construction, and type of glass, but to date it appears to be a safe figure.

Assume an outside condition of 0F and 100% relative humidity with the car at 70F and windows at 14F, inside. As the passengers are the only source of water vapor inside the car, the minimum volume of air required to eliminate fogging can be figured per passenger, the result being multiplied by the number of passengers. Under these conditions, the minimum quantity of outside air to introduce per passenger is 1.53 lb. per min., or 17.8 c.f.m. If the car contains five passengers approximately 90 c.f.m. will be required.

The actual heating load in passenger cars naturally varies with the individual design of car, as well as with speed. Experience shows a fairly safe figure for present day five-passenger enclosed cars to be 160 B.t.u. per hr. per degree difference of inside and outside temperature at 50 m.p.h. with driver only, and excluding outside air leakage. Bus design varies to such an extent that it is

impossible to give even a rough figure covering this. On many bus designs it is not economical to supply sufficient fresh air to exclude leaks, thus adding another variable. In other words, a bus must be carefully figured and should be actually tested for heat loss before specifying equipment.

Assuming a five-passenger car at 50 m.p.h. in zero weather, to be maintained at 70F with air supplied from outside and heated to 140F, the heat load will be 160 x (70-0)F or 11,200 B.t.u. per hr. To deliver this heat, the air will be cooled from 140F to 70F, giving a 70F temperature drop. Figuring the volume of air at inlet conditions, namely 0F, with a specific heat of 0.241 B.t.u. per lb., the weight of air to supply will be:

$$\frac{11{,}200}{(140\text{-}70) \times 0.241 \times 60} = 11 \text{ lb. per min.}$$

As 1 lb. of 0F air contains 11.6 cu. ft. the volume of outside air required will be 11 x 11.6 or 128 c.f.m. Comparing this with the 90 c.f.m. required to eliminate fog, it is obvious that in this case the system must be designed on the basis of heat requirements.

The design of winter equipment for city buses calls for meeting conditions somewhat different than those desired in the private car. City bus passengers are liable to be out in the open for long periods, and so are dressed for the occasion. To maintain such a bus at 70F in zero weather would be inviting a pneumonia epidemic. Fifty degrees in such weather appears to be more desirable, although the best temperature for city bus service at various outside conditions still needs considerable investigation. In city service a 40-passenger bus may vary from no passengers to the full complement, may run with doors opening frequently or infrequently, and may be in the shelter of buildings or on a wind swept stretch, all in the period of half an hour.

The school bus calls for practically the same consideration as the city bus, but unfortunately gets even less. The inter-city bus falls in practically the same category as the private car as far as desired conditions are concerned. Here the passengers can remove their wraps, and door openings are less frequent.

The correct volume of fresh air to supply per passenger for bus ventilation alone, to date is not known, and will probably be the source of considerable argument for some time to come. One transit commission has set an arbitrary figure of 10 c.f.m., which appears to be ample for a city bus, if smoking is not allowed. Experience points to a minimum of 15 c.f.m. per passenger in private cars with passengers smoking. It would, therefore, seem safe to take this figure as a maximum required in any bus for ventilation alone. The volume of air required to eliminate fogging of windows is a definite quantity for a given set of conditions, and is figured in exactly the same manner as for the passsenger car. As in the passenger car, it will be found that if this volume of fresh air is introduced into the bus, ventilation will be ample to satisfy the customers. Considering the number of school bus accidents attributed to fogged windows, it would appear as though this method of figuring ventilation requirements should be used from a safety standpoint alone.

The problem of maintaining proper conditions inside a vehicle in summer is approached in the same fundamental manner as the winter problem, except that heat is removed rather than added, and that proper conditions are affected by different factors. In winter, clear vision was felt to be more desirable than maintaining proper humidity for comfort. In summer, clear vision will not be affected by maintaining proper comfort conditions, and therefore the amount of humidity desired will be based upon comfort alone. Once the desired dewpoint, heat gains, and temperatures are decided, the problem of how much air to deliver to the passenger space is figured by the same method as was used in the winter problem. Since the dewpoint of the air entering is under the designer's control in the summer and not fixed as in the winter problem, it is generally necessary to make several trial solutions before arriving at an economical design. Also, since the volume of air required for removing heat is far greater than required to remove moisture, the problem calls for a system supplying part fresh and part recirculated air.

Excluding leakage of air from outside into the vehicle, the cooling load inside the body will be due to the difference in inside and outside temperature, effect of the sun, and the passengers. Each passenger will give up 260 B.t.u. per hr. The load due to conduction through the car body in summer cannot be arrived at by a simple constant times the difference between inside and outside air temperature. For winter use, a figure of 160 B.t.u. per hr. per degree difference was given as a safe figure for most five-passenger cars, including the heat delivered through the dash plus the effect of the warm air from the engine surrounding part of the body. In summer the heat from the engine is increasing, rather than decreasing, this load. Also, since winter fronts are discarded and louvers opened wide, a large volume of hot air now surrounds part of the body, thus increasing the average surround temperature. The sun generally heats the roof to a temperature higher than the surrounding air and so this part of the body load cannot be considered as a direct function of inside and outside temperature. If the roof is hotter than the surrounding air, it must be giving up heat to that air, as well as to the inside of the car. Also, as the car speed increases, the amount of heat given up to the outside air by the roof will increase, thus reducing the roof temperature and so reducing the cooling load required for this item. One test showed a drop in roof temperature from 135F at 20 m.p.h. to 115F at 60 m.p.h. with a corresponding 35% reduction of roof load. The floor of the car is subjected to radiant heat from the hot road, as well as hot air from the engine. This load will increase or decrease with speed depending upon the individual design of car. The remaining loads are those due to sun entering windows and warming panels, and conduction due to temperature differences, the latter generally increasing with speed.

Considering these variables, it is obviously impossible to give any safe "rule of thumb" figure for summer heat loads due to conduction and sun effect. Therefore, for purposes of illustration, this load will be taken as found experimentally for one medium priced four-door five-passenger car, namely 6200 B.t.u. per hr. at 95F.

The problem of controlling desired conditions in a motor vehicle is far more complex than in a residence

or public building. The building is stationary, and has comparatively thick walls, plus a large volume of enclosed air at the proper conditions, both of which take care of sudden changes in load. The motor vehicle, on the other hand, is operating at widely varying speeds, with sudden changes in heat load, and practically no stabilizing effect. All the air in the car may be replaced in the space of a minute, an unheard of figure in any building. The walls are of light weight construction, even if insulated, and carry a large percentage of glass, all of which makes the car particularly responsive to changes in outside conditions, especially solar radiation. Under these considerations it is evident that any system of control utilized, whether manual or automatic, must be capable of rapid and positive operation.

Under winter conditions the primary prerequisite of comfort is the inside temperature. Since this temperature is to be maintained by introducing heated fresh air, the simplest method of control is to vary the quantity of hot water passing through the heating coil. As previously mentioned, in city buses it is practically imperative to have this control automatic. In private cars it can be controlled manually, but if continuous comfort is to be sold control should be automatic. It must be remembered in any manually-controlled comfort system, the control is not touched until the occupants become uncomfortable.

Considering the domestic market, it is safe to assume that a conditioning system will not be installed for summer conditions alone, but will be designed for year-round operation. In other words, it will have a heating coil operated by hot water, and a cooling coil operated by a separate refrigerating system, either of which can be operated independently of the other, and both of which are ample to take care of maximum heating or cooling requirements. Such a system is an ideal setup for dewpoint control.

In selecting a refrigerating system for a motor vehicle we again encounter many problems not found in stationary systems. Space is at a premium; vibration may cause breaks in lines; power is available, but at widely varying speeds; occupied space is small, giving rise to serious dangers from leaks; accidents may entrap passengers in a space open to refrigerant vapors. With these considerations in mind, it is fairly safe to assume the safest system will be one using a low-pressure refrigerant.

A great temptation naturally exists to use an absorption, or heat operated system of refrigeration in conjunction with the exhaust of an automobile engine. Although such a system is possible, it will be found to be too bulky and too expensive to be practical, to say nothing of the chemical research necessary. Insufficient capacity will also be encountered under idling conditions.

When gasoline is vaporized it absorbs heat, in the same manner as any other liquid. It has therefore been proposed to utilize this principle for cooling motor vehicles. Assuming a car operating at 30 m.p.h. and running at 10 miles per gal., the best cooling effect possible would be roughly one-third of a ton or about one-third the required rate for comfort. If this system was attempted in conjunction with the standard carburetor by passing the mixture through a cooling coil placed between the carburetor and the engine, the resulting temperature of the mixture would not be low enough to condense moisture out of the air and would therefore be of little benefit. If a fuel such as propane is used instead of gasoline, we will still have roughly one-third the amount of cooling effect required, and it will be found more economical to install a straight refrigerating system.

The most logical system appears to be the use of a compressor, operated from the engine through a suitable variable speed drive. Such a system will require a minimum space and not over two horsepower; a figure well below the power required to operate the present radiator fan.

Just what form this system will take, depends upon future development work. However, it is certain that in the near future some form of safe economical automotive refrigerating system will appear.

## Effect of Altitude on Steam Radiators

A USUALLY neglected factor which affects the heat given off by steam radiators is the altitude. In cities such as Denver, Colorado, (elev. 5280 ft.) Cheyenne, Wyoming, (elev. 6060 ft.) and Las Vegas, New Mexico, (elev. 6714 ft.) steam radiators will give off from 7 to 10% less heat than they will in systems located in sea level cities such as New York, Washington, and Houston, Texas.

EFFECT OF ALTITUDE ON HEAT GIVEN OFF BY STEAM RADIATORS

| Altitude Above Sea Level, Feet | Per Cent of Heat Given Off |
|---|---|
| 0 | 100 |
| 1000 | 99 |
| 2000 | 97 |
| 4000 | 94 |
| 6000 | 92 |
| 8000 | 90 |
| 10000 | 87 |
| 12000 | 85 |

This decrease in heat given off is due to the reduction of the boiling temperature of water. At 2000 ft. water will boil at about 208F instead of 212F; at 5000 ft. at about 202F; and at 10,000 ft. at about 193F. Thus a steam system operating under atmospheric pressure in Las Vegas, New Mexico, will be filled with 199F steam instead of 212F steam and the radiators will be at a lower temperature than normal and will give off less heat. Therefore, more radiator surface will have to be provided or a higher steam pressure carried. To get a 212F radiator temperature in Las Vegas it would be necessary to carry about 3½-lb. gage pressure.

# DUCTS

## •DESIGN

Friction in Straight Ducts

# DUCTS

## • DESIGN

### Friction in Straight Ducts

DUCT design or construction cannot well go forward until the amount of frictional energy necessary to move the air along the duct is known or estimated within reasonable limits. A brief consideration of the cause of the friction will help keep the necessary thinking along right paths.

When air is slightly compressed, as by a fan, confined in a duct system and moved along the ducts, the particles of air next to the surface do not slip along freely. The duct surfaces resist the free movement of the air. This resistance is due to friction. In the same way air particles try to slide over each other and meet resistance in doing so. This resistance is also a frictional resistance.

As a result the air can move along the duct only if urged along by a force at least equal to the opposing frictional resistance. Overcoming this resistance for a period of time in a practical duct system requires the expenditure of energy. One of the important steps in duct design is to attempt to find out in advance of construction how much this expenditure of energy is going to be.

#### What Friction Depends on

It is generally agreed that when air at low pressure moves through ventilating ducts the friction can be best judged by applying the same analysis as used when water flows through pipes. In such cases many investigators have found that the force necessary to overcome duct friction depends on several factors including principally:

(1) The *roughness* of the duct and the degree of *turbulence* of the air flowing, which fix a value on the coefficient of friction $f$; (2) the *length* of the duct $L$; (3) the *speed* of the air in the duct $V$ so that an increase in any of these factors increases the force (and the energy) necessary to overcome friction, while (4) the *diameter* of the duct $D$ decreases the force necessary as the diameter of the duct increases.

This relationship is often set up mathematically as:

$$\text{Friction Loss} = h_f = f\left(\frac{L}{D}\right)\left(\frac{V^2}{2g}\right) \quad \ldots\ldots\ldots\ldots (1)$$

This equation has been subjected to much mathematical treatment and transformation in order to get it into a form which is useful for specific purposes. Also the equation has been the basis of a great deal of experimental work as various investigators have attempted to set up numerical values for the factors or to check those already established by others. As a result there are now prevalent many modifications of this general formula and a considerable variety of numerical values for use in the equation. Also there is some difference of opinion as to certain points connected with the general theory back of the equation but these need not be discussed here.

It is sufficient to note for present purposes that the application of this equation can be accurate and trustworthy in predicting the frictional energy only as the values placed on the coefficient of friction $f$ are the correct ones. These values have to be found by experiment or experience, are not particularly easy to measure and consequently the findings reported are not uniform or consistent. As a result predictions of frictional losses in air ducts for design purposes must still be looked on as being estimates. So generally is this recognized that the designer virtually always protects himself by making provision in his designs for dampers or similar adjustable devices to vary the resistance and thus take care of unpredictable conditions as they arise. The search for correct values of $f$ has gone on for a long period along several different fronts, but in the meantime use has to be made of the formula. A number of people have looked into the situation but have seldom agreed as to just what is to be considered the best and most desirable values of $f$ to use.

#### Friction Charts

Also when the question of usefulness of the equation has been considered the usual conclusion has been that some form of chart should be plotted. With various people devising charts and with no specific agreement as to the basic values to be used it is not surprising that a variety of working charts has gradually come into existence.

Obviously it is possible to prepare charts in a great many ways, of various sizes, and covering any desired range of capacity and air speed. Perhaps the most widely printed chart uses double logarithmic scales on rectangular coordinates and gives a network of sensibly straight lines criss-crossing the chart. Charts of this general form have been in existence for a long period and widely used. The one presented here as Fig. 1 is a plot of equation (1) modified to give effect to a variation in the value of the coefficient of friction as reported by certain experimenters who found that the relation is more correctly stated if the equation takes the form

$$\text{Friction Loss} = h_f = 1.1\,\frac{L}{CD^{5/3}}\left(\frac{V}{4005}\right)^{13/7} \quad \ldots\ldots\ldots\ldots (2)$$

*(Text concluded on page 48)*

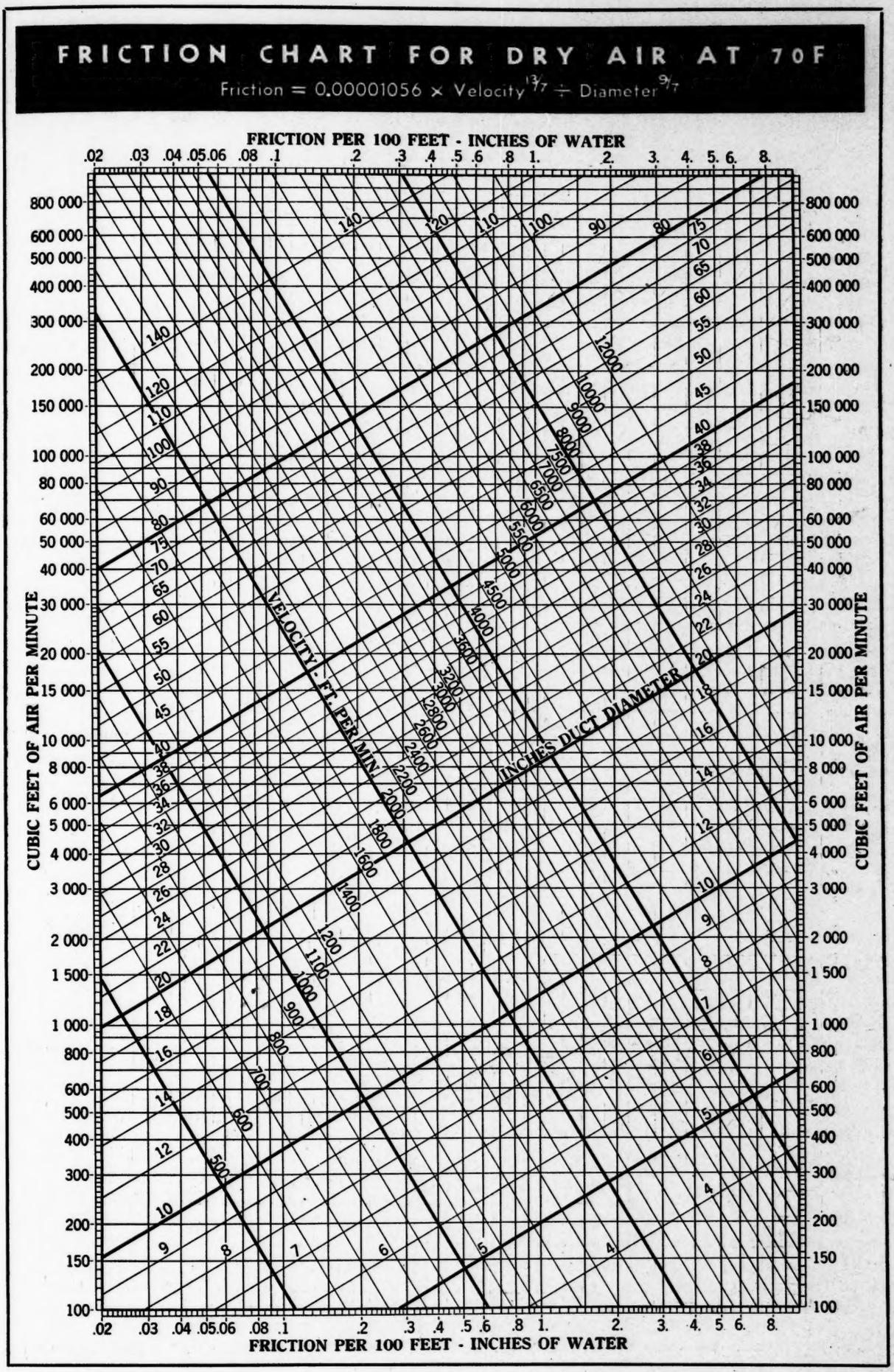

FRICTION CHART FOR DRY AIR AT 70F
Friction = 0.00001056 × Velocity^13/7 ÷ Diameter^9/7
FRICTION PER 100 FEET - INCHES OF WATER
CUBIC FEET OF AIR PER MINUTE
VELOCITY - FT. PER MIN.
INCHES DUCT DIAMETER

*Fig. 1*

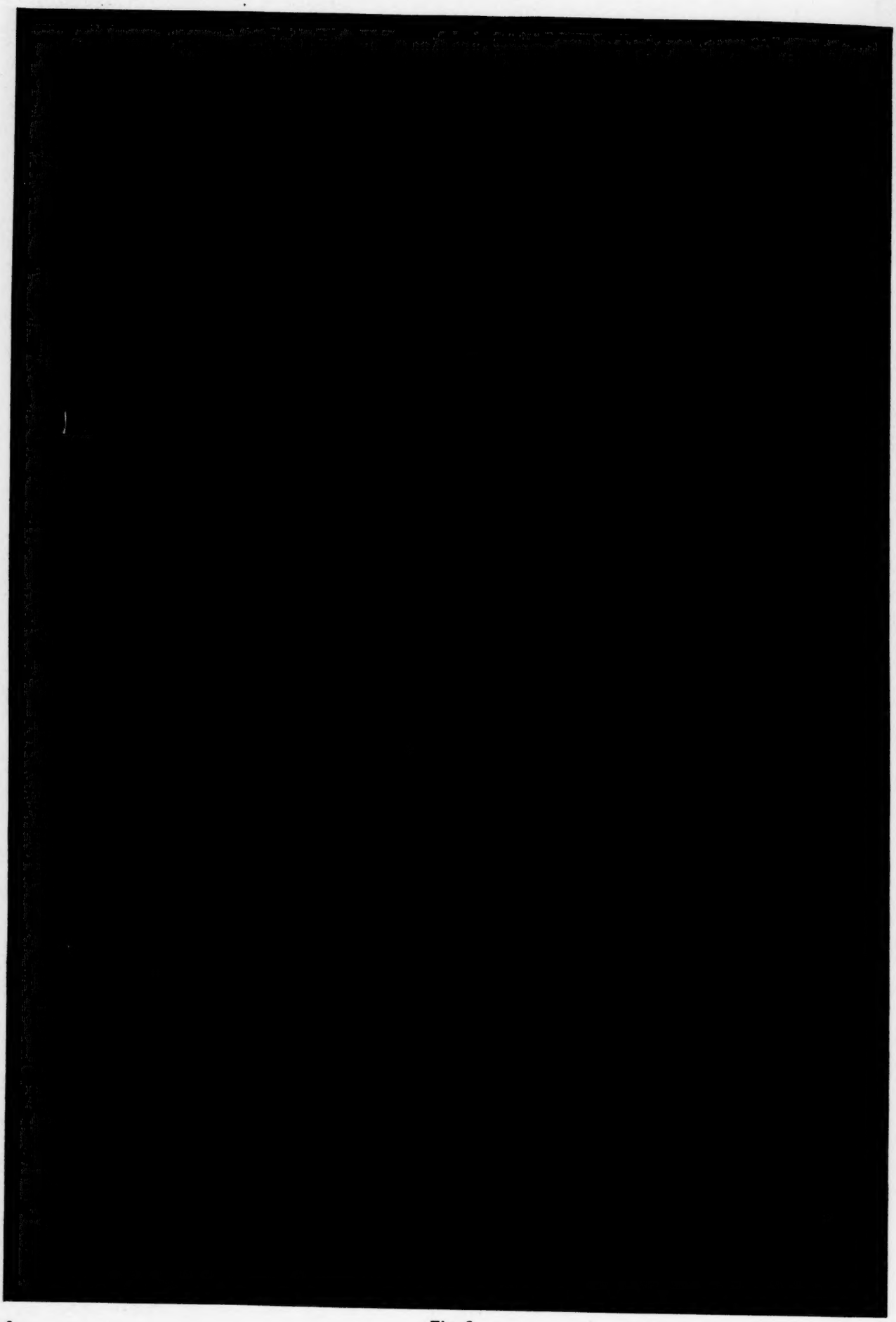


*Fig. 2*

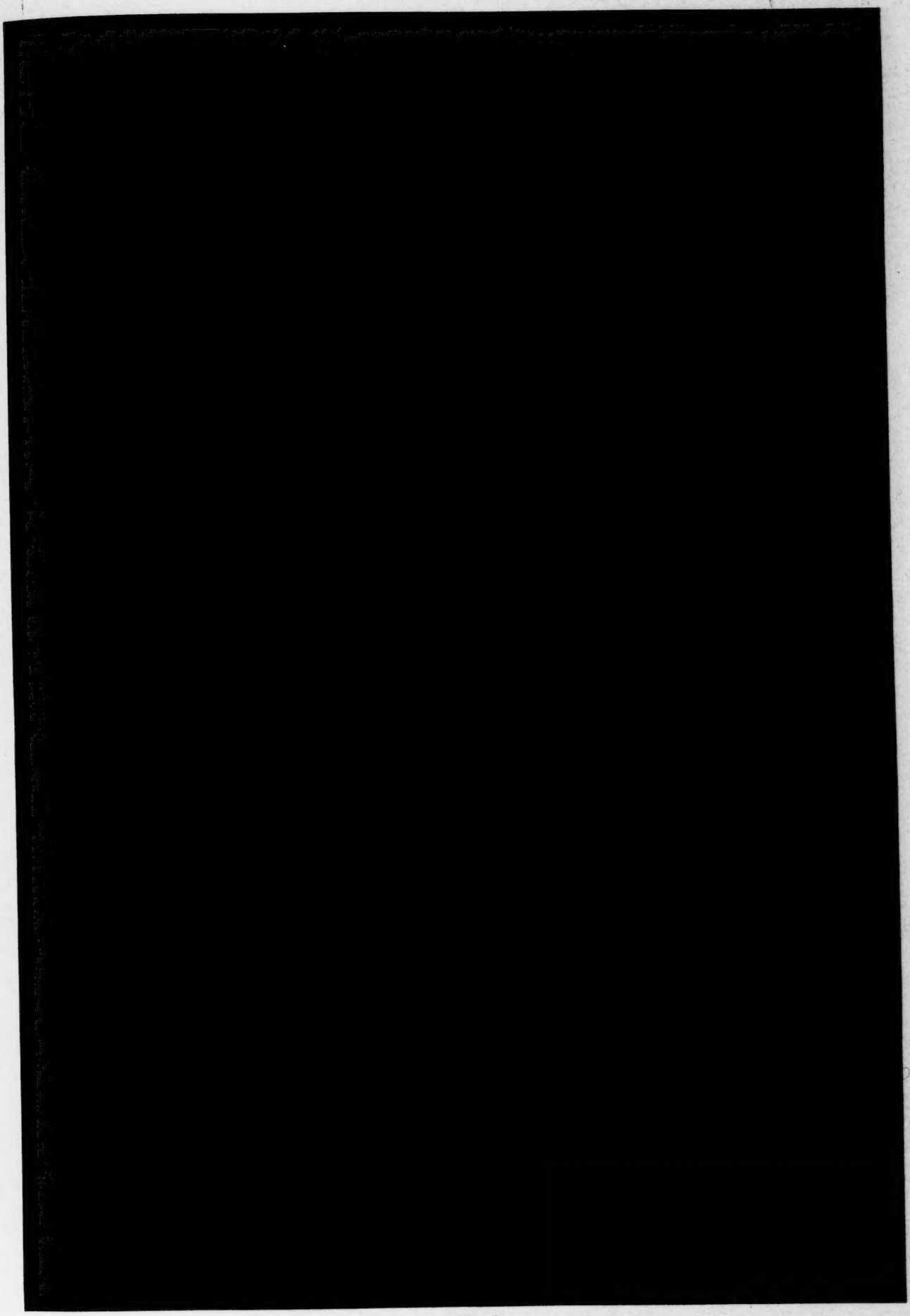


*Fig. 3*

# FRICTION CHART FOR 130F AIR AT 45F DEWPOINT

Friction = 0.000268 × f × Velocity$^2$ ÷ Diameter,
where f = a function of the Reynolds number

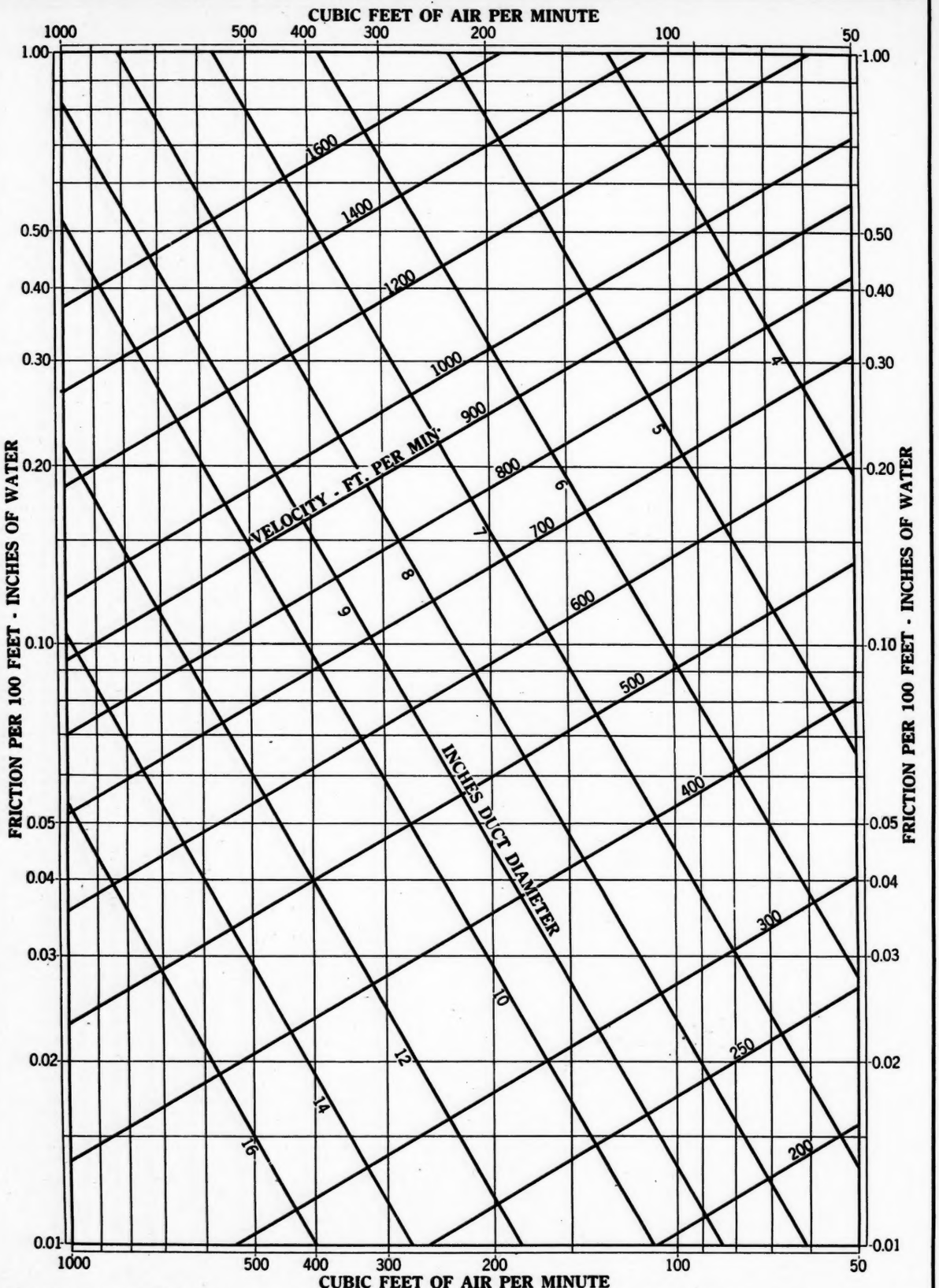



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*Fig. 4*

## TABLE 1—EQUIVALENT CIRCULAR AND RECTANGULAR DUCTS WITH EQUAL FRICTION

| One Side Rect'lar Duct, Inches ↓ | Other Side of Rectangular Duct, Inches | | | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 14 | 16 | 18 | 20 | 22 | 24 | 26 | 28 | 10 | 32 | 34 | 36 | 38 | 40 | 42 | 44 | 46 | 48 |
| | Diameter of Circular Duct, Inches | | | | | | | | | | | | | | | | | | | | | | | | | | | |
| 3½ | — | — | — | — | — | 5.7 | 6.0 | 6.3 | 6.6 | 6.9 | 7.4 | 7.8 | | | | | | | | | | | | | | | | |
| 4 | 3.8 | 4.4 | 4.9 | 5.4 | 5.8 | 6.1 | 6.5 | 6.8 | 7.1 | 7.4 | 7.9 | 8.4 | | | | | | | | | | | | | | | | |
| 4½ | 4.0 | 4.7 | 5.2 | 5.8 | 6.1 | 6.5 | 6.9 | 7.2 | 7.6 | 7.9 | 8.5 | 9.0 | | | | | | | | | | | | | | | | |
| 5 | 4.3 | 4.9 | 5.5 | 6.3 | 6.5 | 6.9 | 7.3 | 7.7 | 8.0 | 8.3 | 8.9 | 9.5 | | | | | | | | | | | | | | | | |
| 5½ | 4.4 | 5.2 | 5.9 | 6.4 | 6.8 | 7.3 | 7.7 | 8.1 | 8.5 | 8.8 | 9.5 | 10.1 | | | | | | | | | | | | | | | | |
| 6 | 4.6 | 5.4 | 6.3 | 6.6 | 7.0 | 7.6 | 8.0 | 8.4 | 8.8 | 9.2 | 9.9 | 10.5 | | | | | | | | | | | | | | | | |
| 7 | 5.0 | 5.8 | 6.5 | 7.0 | 7.7 | 8.2 | 8.7 | 9.2 | 9.6 | 10.0 | 10.8 | 11.4 | | | | | | | | | | | | | | | | |
| 8 | 5.3 | 6.1 | 6.9 | 7.6 | 8.2 | 8.8 | 9.3 | 9.8 | 10.2 | 10.7 | 11.5 | 12.3 | | | | | | | | | | | | | | | | |
| 9 | 5.5 | 6.5 | 7.3 | 8.0 | 8.7 | 9.3 | 9.9 | 10.4 | 10.9 | 11.4 | 12.3 | 13.1 | | | | | | | | | | | | | | | | |
| 10 | 5.8 | 6.8 | 7.7 | 8.4 | 9.2 | 9.8 | 10.4 | 11.0 | 11.5 | 12.0 | 12.9 | 13.8 | | | | | | | | | | | | | | | | |
| 11 | 6.1 | 7.1 | 8.0 | 8.8 | 9.6 | 10.2 | 10.9 | 11.5 | 12.1 | 12.6 | 13.6 | 14.5 | | | | | | | | | | | | | | | | |
| 12 | 6.3 | 7.4 | 8.3 | 9.2 | 10.0 | 10.7 | 11.4 | 12.0 | 12.6 | 13.2 | 14.3 | 15.2 | | | | | | | | | | | | | | | | |
| 13 | 6.4 | 7.6 | 8.7 | 9.6 | 10.4 | 11.1 | 11.8 | 12.5 | 13.1 | 13.7 | 14.8 | 15.8 | | | | | | | | | | | | | | | | |
| 14 | 6.7 | 7.9 | 8.9 | 9.9 | 10.8 | 11.5 | 12.3 | 12.9 | 13.6 | 14.3 | 15.4 | 16.5 | | | | | | | | | | | | | | | | |
| 15 | 7.0 | 8.2 | 9.2 | 10.2 | 11.1 | 11.9 | 12.7 | 13.4 | 14.1 | 14.7 | 16.0 | 17.1 | | | | | | | | | | | | | | | | |
| 16 | 7.2 | 8.4 | 9.5 | 10.5 | 11.4 | 12.3 | 13.1 | 13.8 | 14.5 | 15.2 | 16.5 | 17.6 | | | | | | | | | | | | | | | | |
| 17 | 7.3 | 8.6 | 9.8 | 10.8 | 11.8 | 12.6 | 13.5 | 14.2 | 15.0 | 15.7 | 17.0 | 18.2 | | | | | | | | | | | | | | | | |
| 18 | 7.5 | 8.9 | 10.0 | 11.1 | 12.1 | 13.0 | 13.8 | 14.6 | 15.4 | 16.1 | 17.4 | 18.7 | 19.8 | | | | | | | | | | | | | | | |
| 19 | 7.7 | 9.1 | 10.3 | 11.4 | 12.4 | 13.3 | 14.2 | 15.0 | 15.8 | 16.5 | 17.9 | 19.2 | 20.4 | | | | | | | | | | | | | | | |
| 20 | 7.9 | 9.3 | 10.5 | 11.6 | 12.7 | 13.6 | 14.5 | 15.4 | 16.2 | 17.0 | 18.4 | 19.7 | 20.9 | 22.0 | | | | | | | | | | | | | | |
| 22 | 8.2 | 9.7 | 11.0 | 12.1 | 13.2 | 14.2 | 15.2 | 16.1 | 16.9 | 17.8 | 19.2 | 20.6 | 21.9 | 23.1 | 24.2 | | | | | | | | | | | | | |
| 24 | 8.5 | 10.0 | 11.4 | 12.6 | 13.8 | 14.8 | 15.8 | 16.8 | 17.6 | 18.5 | 20.0 | 21.5 | 22.8 | 24.0 | 25.2 | 26.4 | | | | | | | | | | | | |
| 26 | 8.8 | 10.4 | 11.8 | 13.1 | 14.3 | 15.4 | 16.4 | 17.3 | 18.3 | 19.2 | 20.8 | 22.3 | 23.8 | 25.1 | 26.3 | 27.5 | 28.6 | | | | | | | | | | | |
| 28 | 9.1 | 10.8 | 12.2 | 13.5 | 14.8 | 15.9 | 17.0 | 18.0 | 19.0 | 19.8 | 21.5 | 23.1 | 24.6 | 26.0 | 27.3 | 28.5 | 29.7 | 30.8 | | | | | | | | | | |
| 30 | 9.2 | 11.0 | 12.6 | 13.9 | 15.2 | 16.4 | 17.5 | 18.5 | 19.5 | 20.5 | 22.2 | 23.9 | 25.4 | 26.8 | 28.2 | 29.5 | 30.7 | 31.9 | 33.0 | | | | | | | | | |
| 32 | 9.6 | 11.3 | 12.9 | 14.3 | 15.6 | 16.9 | 18.0 | 19.1 | 20.1 | 21.1 | 22.9 | 24.6 | 26.2 | 27.7 | 29.1 | 30.5 | 31.7 | 32.9 | 34.1 | 35.2 | | | | | | | | |
| 34 | 9.9 | 11.6 | 13.2 | 14.7 | 16.1 | 17.3 | 18.5 | 19.6 | 20.7 | 21.6 | 23.5 | 25.3 | 26.9 | 28.5 | 30.0 | 31.3 | 32.7 | 33.9 | 35.1 | 36.3 | 37.4 | | | | | | | |
| 36 | 10.0 | 11.9 | 13.6 | 15.1 | 16.4 | 17.7 | 19.0 | 20.1 | 21.2 | 22.2 | 24.2 | 26.0 | 27.7 | 29.3 | 30.8 | 32.2 | 33.7 | 34.9 | 36.1 | 37.3 | 38.5 | 39.6 | | | | | | |
| 38 | — | 12.2 | 13.9 | 15.4 | 16.8 | 18.2 | 19.4 | 20.6 | 21.7 | 22.8 | 24.8 | 26.7 | 28.4 | 30.0 | 31.5 | 33.1 | 34.6 | 35.9 | 37.1 | 38.4 | 39.5 | 40.7 | 41.8 | | | | | |
| 40 | — | 12.5 | 14.3 | 15.7 | 17.2 | 18.6 | 19.8 | 21.1 | 22.2 | 23.3 | 25.4 | 27.3 | 29.1 | 30.8 | 32.4 | 33.9 | 35.3 | 36.7 | 38.0 | 39.3 | 40.5 | 41.7 | 42.9 | 44.0 | | | | |
| 42 | — | 12.7 | 14.5 | 16.1 | 17.6 | 19.0 | 20.3 | 21.6 | 22.7 | 23.8 | 25.9 | 27.9 | 29.8 | 31.4 | 33.0 | 34.5 | 36.0 | 37.6 | 39.0 | 40.3 | 41.5 | 42.7 | 44.0 | 45.1 | 46.2 | | | |
| 44 | — | 13.0 | 14.8 | 16.4 | 18.0 | 19.4 | 20.7 | 22.0 | 23.1 | 24.3 | 26.5 | 28.5 | 30.3 | 32.1 | 33.7 | 35.3 | 36.9 | 38.5 | 39.9 | 41.2 | 42.5 | 43.7 | 44.9 | 46.1 | 47.2 | 48.4 | | |
| 46 | — | 13.3 | 15.1 | 16.7 | 18.4 | 19.8 | 21.1 | 22.4 | 23.6 | 24.8 | 27.0 | 29.1 | 31.0 | 32.8 | 34.6 | 36.2 | 37.8 | 39.3 | 40.8 | 42.2 | 43.5 | 44.8 | 46.0 | 47.2 | 48.4 | 49.5 | 50.6 | |
| 48 | — | 13.5 | 15.4 | 17.0 | 18.7 | 20.1 | 21.5 | 22.8 | 24.1 | 25.2 | 27.5 | 29.6 | 31.6 | 33.4 | 35.2 | 37.0 | 38.5 | 40.0 | 41.5 | 43.0 | 44.4 | 45.6 | 46.9 | 48.1 | 49.3 | 50.5 | 51.6 | 52.8 |
| 50 | — | 13.7 | 15.7 | 17.3 | 19.0 | 20.4 | 21.9 | 23.2 | 24.5 | 25.7 | 28.0 | 30.3 | 32.2 | 34.1 | 35.9 | 37.6 | 39.2 | 40.8 | 42.3 | 43.8 | 45.2 | 46.5 | 47.9 | 49.1 | 50.4 | 51.6 | 52.9 | 54.0 |
| 52 | — | 13.9 | 15.9 | 17.6 | 19.2 | 20.8 | 22.2 | 23.6 | 24.9 | 26.2 | 28.5 | 30.7 | 32.9 | 34.7 | 36.5 | 38.3 | 40.0 | 41.6 | 43.1 | 44.7 | 46.1 | 47.5 | 48.9 | 50.1 | 51.3 | 52.5 | 53.8 | 55.0 |
| 54 | — | 14.1 | 16.1 | 17.9 | 19.6 | 21.1 | 22.6 | 24.0 | 25.3 | 26.6 | 29.0 | 31.2 | 33.4 | 35.3 | 37.2 | 38.9 | 40.7 | 42.4 | 44.0 | 45.5 | 47.0 | 48.4 | 49.9 | 51.1 | 52.3 | 53.5 | 54.8 | 56.0 |
| 56 | — | 14.3 | 16.3 | 18.2 | 19.9 | 21.5 | 22.9 | 24.4 | 25.7 | 27.0 | 29.5 | 31.7 | 33.9 | 35.9 | 37.8 | 39.6 | 41.3 | 43.0 | 44.6 | 46.2 | 47.7 | 49.1 | 50.6 | 52.0 | 53.3 | 54.6 | 55.9 | 57.0 |
| 58 | — | 14.6 | 16.6 | 18.4 | 20.2 | 21.8 | 23.3 | 24.7 | 26.1 | 27.4 | 30.0 | 32.2 | 34.4 | 36.4 | 38.4 | 40.3 | 42.1 | 43.8 | 45.4 | 47.0 | 48.5 | 50.0 | 51.5 | 52.9 | 54.2 | 55.5 | 56.8 | 58.0 |
| 60 | — | 14.7 | 16.8 | 18.7 | 20.4 | 22.1 | 23.6 | 25.1 | 26.5 | 27.8 | 30.5 | 32.7 | 34.9 | 37.1 | 39.1 | 40.9 | 42.7 | 44.5 | 46.1 | 47.8 | 49.3 | 50.9 | 52.3 | 53.8 | 55.0 | 56.4 | 57.7 | 58.9 |
| 62 | — | 15.0 | 17.0 | 19.0 | 20.7 | 22.4 | 24.0 | 25.5 | 26.9 | 28.2 | 30.9 | 33.2 | 35.4 | 37.7 | 39.6 | 41.6 | 43.4 | 45.1 | 46.8 | 48.4 | 50.0 | 51.7 | 53.0 | 54.5 | 55.9 | 57.2 | 58.5 | 59.7 |
| 64 | — | 15.1 | 17.3 | 19.2 | 21.0 | 22.7 | 24.3 | 25.9 | 27.3 | 28.6 | 31.3 | 33.7 | 35.9 | 38.2 | 40.2 | 42.2 | 44.0 | 45.8 | 47.5 | 49.2 | 50.9 | 52.4 | 53.9 | 55.4 | 56.8 | 58.1 | 59.4 | 60.6 |
| 66 | — | 15.3 | 17.5 | 19.5 | 21.2 | 23.0 | 24.6 | 26.2 | 27.7 | 29.0 | 31.7 | 34.2 | 36.4 | 38.7 | 40.8 | 42.8 | 44.7 | 46.5 | 48.2 | 50.0 | 51.6 | 53.1 | 54.7 | 56.2 | 57.6 | 59.1 | 60.4 | 61.6 |

$$d = 1.265 \sqrt[5]{\frac{(ab)^3}{a+b}}$$

where
$d$ = diameter, in inches, of a round duct to produce equal friction per foot of length, when carrying the same amount of air as a rectangular duct $a$ inches by $b$ inches in dimension.

The chart Fig. 1 is plotted by substituting 100 for $L$, 50 for $C$, and changing the units to inches of duct diameter rather than feet. These changes cause the equation to become

$$h_{100} = 0.00001056 \frac{V^{12/7}}{d^{9/7}} \quad \text{.....(3)}$$

where $V$ is in feet per minute and $d$ is in inches of round duct diameter. The units in the equation also make the chart apply to dry air at 70F. It should also be noted that the chart of Fig. 1 is based on a loss of one velocity head at a velocity of 2000 f.p.m. in a 24-in. duct.

From this it will be seen that Fig. 1 is laid out for a particular set of conditions and that it is not a universal graph of the relation between friction, air speed, and duct diameter. It has been widely used, nevertheless, and is probably the most frequently printed friction chart for air flowing in ducts, since it is found in many text and reference books.

Both scales of Fig. 1 extend to limits seldom encountered in ordinary ventilating and air conditioning work. To bring the limits into closer agreement with these conditions Fig. 2 has been prepared. It is a wholly new chart not heretofore published. It is plotted from the equation

$$h_{100} = 0.0000105 \frac{V^{1.86}}{d^{1.26}} \quad \text{.....(4)}$$

This chart is also based on dry air at 70F. It has the advantage over Fig. 1 of being so plotted that the intervals are larger, or so that it can frequently be read closer than can Fig. 1 on the most usually encountered work without enlarging the chart unduly.

As ducts are frequently designed to carry air at temperatures other than 70F and containing a considerable degree of water vapor, Fig. 3—another wholly new chart—has been prepared by modifying Fig. 2 to give effect to the changes brought about when 170F air at 45 dewpoint is being handled.

With these three charts available the user can estimate his duct friction losses in round ducts to meet the conditions encountered in most cases. Where the conditions in the ducts are greatly different from either dry air at 70F or moist air at 170F, the situation can be met by keeping in mind that the friction varies directly as the air density and nearly inversely as the absolute temperature.

As all the charts are essentially similar a single example of their use will suffice for all.

*Example:* Assume that it is necessary to pass 1200 c.f.m. at 170F and 45F dewpoint through 80 ft. of straight 12-in. diameter round duct (a) What will be the friction loss in inches of water? (b) What would the friction loss be if the air were dry at 70F?

*Solution:* On Fig. 3 locate 1200 c.f.m. on the vertical scale, move horizontally until the diagonal line marked 12 in. is intersected; the other diagonal shows that the air will move at 1525 f.p.m. From the intersection move vertically up or down to the horizontal scale and find that the friction is 0.26 in. of water per 100 ft. For 80 ft. the loss is $0.80 \times 0.26 = 0.208$ in. of water. To find the answer (b) use either Fig. 1 or Fig. 2 and proceed similarly. It will be found that the friction is 0.36 in. per 100 ft. or 0.288 in. in 80 ft.

## Friction in Rectangular Ducts

Charts similar to Figs. 1, 2, and 3 can be constructed for determining the friction in rectangular ducts. An alternate method is to prepare a table showing the dimensions of rectangular ducts which produce equal friction losses when carrying the same quantity of air as does a circular duct. With such a table the equivalent round ducts can be found and the charts for round ducts can be used directly. Table 1 is presented for this purpose. It covers the range of rectangular ducts usually met with. In those cases where the duct is not included the equivalent diameter can be found from the equation,

$$d = 1.265 \sqrt[5]{\frac{(ab)^3}{a+b}} \quad \text{.....(5)}$$

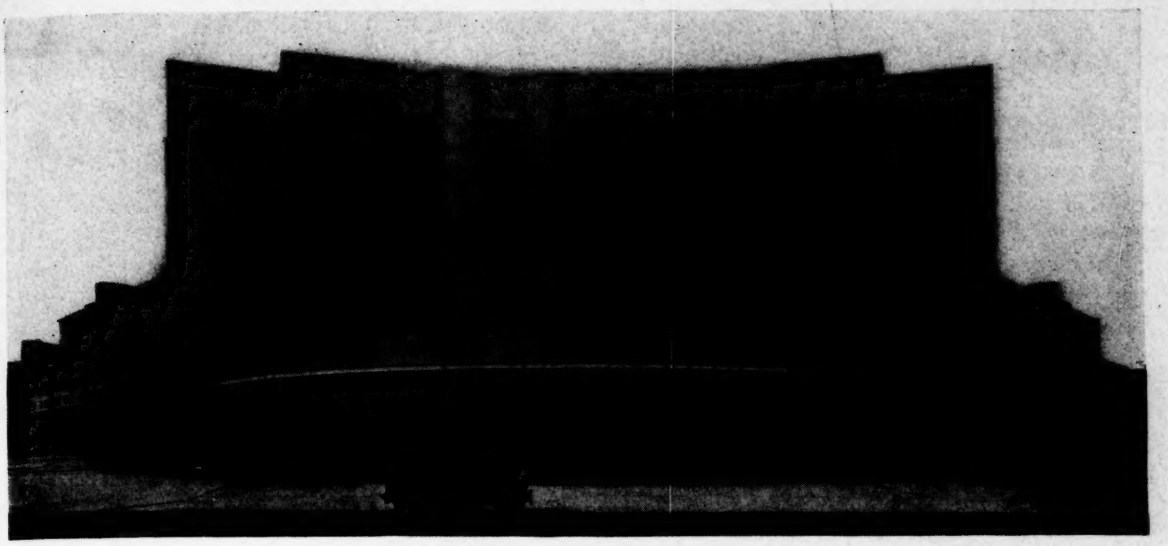
where $d$ = diameter in inches of a round duct to produce equal friction per foot of length when carrying the same amount of air as a rectangular duct having dimensions $a$ inches on one side and $b$ inches on the other.

There has frequently been criticism of friction charts plotted in the manner of Figs. 1, 2, and 3, and all like charts. Objections to their use have been based on either of two general grounds. One is that the value of the friction factor $f$ is not properly set, and the other is that the experimental information on which the charts are based is not conclusive or reliable.

So far as the second objection is concerned it is correct to conclude that the charts as presented are based on the best experimental information available. If this information is not reliable, nothing can be done about it until definite evidence is advanced to either prove or disprove beyond question the correctness of the contention.

Methods of evaluating the friction factor $f$ have frequently been studied and reported by many writers. The possibility and advisability of constructing friction charts based on evaluating the friction factor by the use of Reynold's numbers have been presented repeatedly both in HEATING & VENTILATING and elsewhere. Few such friction charts have actually been made available, however. One chart of this kind covering a limited range is presented in Fig. 4. It also is based on the best experimental evidence at hand. If this chart is used, somewhat different duct dimensions may result than if Fig. 3, for example, is used as the guide. At present the user must judge for himself as to whether he should base his estimates on charts plotted in the manner of Figs. 1 to 3 or on the limited range chart of Fig. 4.

Estimates of friction losses in duct systems are readily and easily made by the discriminating and careful use of the charts and tables presented here but it is worthy of note that there is not full agreement among competent engineers as to how closely the operating plant when built will come to meeting the frictional losses estimated before construction. It is a situation in which there is a wealth of detailed procedural information available but whose factual value is difficult to evaluate. Duct friction is usually a matter of such importance that care should be taken to be sure that the estimate is made as accurately as conditions and existing data will permit.



# 2000 Tons of Air per Hour for Ventilation

THE new Earls Court Exhibition Center in London, England, which is claimed to be the largest individual building in the world, has a complete winter air conditioning system with heat provided by central electric heaters. This London building is four stories high, covers an area of 12 acres, and has a volume of 47,000,000 cu. ft. It was erected at a cost of approximately $7½ million.

The shape of the building is roughly triangular, each of the three sides measuring 700 ft. The main floor is 15 ft. above the natural ground level, which has enabled the surrounding roadways to be raised to the upper level of a double-decked car park designed to take 2000 cars. The roof over the central area has a span of 250 ft. x 350 ft. and a height from floor level of 120 ft. Over 20,000 spectators can be accommodated around the arena, which can be used as an exposition center or a swimming pool. By hydraulic means the floor of the swimming pool can be brought up to the main floor in three sections.

In general, plenum systems of ventilating are employed, there being in all over 90 complete installations, comprising fans, motors, and ductwork, which in the aggregate are rated to handle about 2000 tons of air per hour. All the necessary heat is derived from an electric thermal storage plant.

Heating is carried out by two methods—direct heating by means of radiators and air heating in the process of ventilation. The direct heating of the building is not so large as that produced by the plenum system, but is by no means a negligible quantity, for there are 368 radiators installed possessing a total heating surface of 20,000 sq. ft., and the total run of piping used for this process alone is over six miles in length.

**Over 2000 tons of conditioned air are supplied each hour to London's new Earls Court Exhibition Center by an unusual winter air conditioning system. Heat is generated by a central electric hot water plant and is distributed both by radiators and forced warm air. Because of the character of the building and the large volumes of some of the exhibition spaces many unusual methods of air distribution had to be employed. The building has a total volume of 47 million cu. ft. and is said to be the largest individual building in the world.**

Air is treated in the plenum plants by being passed through a water spray for washing and then over a hot water coil for heating.

Hot water is supplied by seven large thermal tanks. The water is electrically heated in boilers to 270F. The stored tank water is mixed with the water in the pipe circuits, so that the temperature of water entering the coils is 180F. The leaving temperature is about 150F.

There are 11 separate pipe circuits from the thermal storage chamber to the various sections of the building. Each circuit employs mixing valves to adjust the water temperature to suit the requirements in different sections of the building.

To ventilate each of the restaurants and swimming pool dressing rooms, plenum plants are installed in the basement. They deliver washed fresh air. The used air is exhausted by another plant, thus producing and maintaining a balanced system of ventilation throughout this section.

The method of ventilating the basement is interesting because of the extensive storage spaces. The exhaust system extracts two changes of air per hour, and has duplicate fans. Each fan set is capable of handling this volume without the help of the other.

An elaborate system of light-actuated devices is placed in various parts of the storage spaces so that when smoke appears in any section, the second fan set automatically starts up and the exhaust system will then extract four changes of air hourly, expelling the smoke at roof level.

Due to the risk of fire in this section, the whole of the distributing ductwork in the storage spaces is constructed throughout in fire-resisting material, and separate ducts are taken from each space right back to the

fans to insure the smoke being removed from its source without danger of spreading to the remaining stores.

The convention and other halls and the restaurants on the mezzanine floors also have separate installations for distributing fresh air and separate balanced plants for extracting used air.

In the main hall, the most interesting part of the entire undertaking, the difficulties to be overcome were:

1. The problem of ventilating and heating a hall 350 ft. wide x 120 ft. high and seating 20,000 persons
2. The installation of a system which would give correct air distribution, (a) both when the hall is being used by a seated audience, and (b) when the seating is removed and the hall becomes an open hall for exposition use.
3. The elimination of noise caused by high velocity nozzles when discharging air into the hall.

In order to observe London County Council requirements, the plenum installation for this hall is designed to provide 1000 c.f.h. per seat. The exhaust system removes only 90% of the air being introduced in order to create a slight pressure.

The main hall installation consists of 8 complete plenum and exhaust plants arranged in chambers running the length of the two sides and one end of the hall at the roof level. The parts of these chambers not actually used as plant rooms are utilized as ducts. Apertures are made in the floors of these ducts for metal discharge nozzles of high velocity type. Six of these nozzles are arranged in each chamber, each discharging 8000 c.f.m. at a fixed velocity and at a predetermined angle pointing towards the center of the hall—a distance of about 145 ft.

This arrangement of air distribution is used when the main hall is adapted for a seated audience.

When the hall is employed for exposition purposes, however, the system uses ducts which connect into the side of the plenum chamber. These ducts descend vertically downward through the rear of the gallery seating, under the raking seats, and then drop to high level on the ground floor, where they are fitted with directional outlet grilles. Similar grilles having high velocity and directional features are fitted to the sides of the ducts under the gallery rakers, and louvered outlets of low velocity are employed at the rear of the gallery.

By means of adjustable metal louvers the air flow through these ducts is controlled at roof level. They are interconnected in such a manner that they can be employed either as plenum ducts when the hall is used for exposition purposes or exhaust ducts when operated in conjunction with the high velocity nozzles if the hall is adapted for a seated audience.

Recirculation louvers and exhaust louvers are also fitted at roof level to enable recirculation or extraction to be performed from the ceiling grilles.

Each plenum plant has an air washer and air heater. In order to economize in heating, the system has been designed to give total recirculation during warming-up periods.

The four other halls are provided with ventilation in a similar manner by using high velocity nozzles, but with no reversible system as in the main hall, the exhaust air being extracted through ceiling grilles, and recirculation being permitted for warming-up.

Air conditioning of the three large convention halls located on the upper floors at the three entrances to the building have to conform with London theater regulations, as they each contain permanent stages. They can hold seated audiences of 550, 870, and 1000 persons, respectively. They are, therefore, provided with full plenum and exhaust systems. In each case the plants consist of an air washer and air heater, together with plenum and exhaust fans.

Four large restaurants—two at mezzanine floor level and two in the basement—have a total seating accommodation for 6000 persons. A full plenum and exhaust system is designed for each on the basis of 1000 c.f.h. per seat for the mezzanine restaurants, and 1500 c.f.h. per seat for those in the basement. Separate plants are provided in each case as well as with separate exhaust apparatus for the service spaces.

The two kitchens in the basement are supplied with air at a rate of 15 air changes per hour, the exhaust being 20 air changes per hour. The plenum plants for these kitchens are fitted with specially designed dry filters of the cellular type.

The remarkable nature of the entire installation may best be appreciated by a statement of selected facts.

The job has been carried out in a building covering 12 acres and containing a capacity of 47,000 cu. ft.

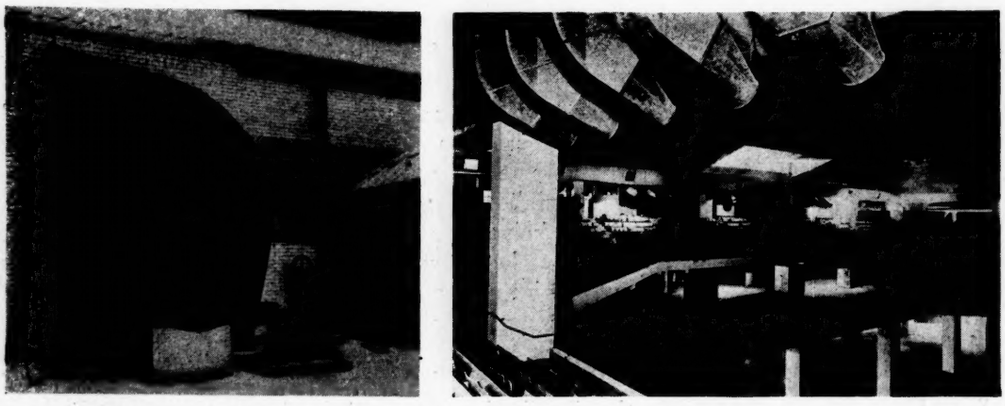
The supply of air necessary to deal with the building amounts to 2000 tons per hr.

The plant undertaking this work consists of 90 complete installations embracing fans, motors, and duct-work.

The total run of piping serving radiators and plenum plants amounts to over nine miles.

The heating piping surface covered with insulation represents a total area of one and a half acres.

The total heat requirements are 46,500,000 B.t.u. per hr.



*(Left) Two fans used to ventilate main convention hall. (Right) A bank of five injector nozzles which supply 40,000 c.f.m. of air to the center of the main floor. This air is projected through a distance of 145 ft.*

# Elimination of Algae in Cooling Towers

**Atmospheric cooling devices used to cool water which is recirculated continuously often become clogged with organic growths or algae. Unless the water is treated to remove this growth the capacity of the device will be materially reduced. One method of treating the water with chlorine gas was recently reported by T. C. Wallace of the Southern Counties Gas Co., California, before the annual convention of the Pacific Coast Gas Association. A description of this method follows.**

ALTHOUGH it has been known for some time that chlorine gas under certain circumstances will destroy the organic growth, or algae, common to most cooling towers, the conditions under which it may be used most effectively for this purpose are quite vague. It has been found that chlorine may readily control the growth of algae in one locality and not in another, conditions apparently being the same at both places.

This company became interested in the use of chlorine for this purpose as the result of a paper presented on the subject by Hugh Multer of the Union Oil Company of California. We are indebted to him for his cooperation and numerous suggestions. This report simply details our own experiences in the use of chlorine to date and should perhaps be considered as a progress report.

Two factors mainly control the use of chlorine gas in cooling tower systems. First, the algae possess the peculiar faculty of being able to immunize themselves against chlorine if the concentration is not strong enough to kill them at once. Second, chlorine is very corrosive towards most metals encountered in compressor plant piping, and this prevents its continued use at high concentrations.

The secret of success in this method of treatment appears to be in injecting suddenly at intervals of once or twice a week, sufficient chlorine to kill the algae. This sudden dosage is strong enough to kill the algae, and since it becomes dissipated to the atmosphere in a few hours it does not attack the piping.

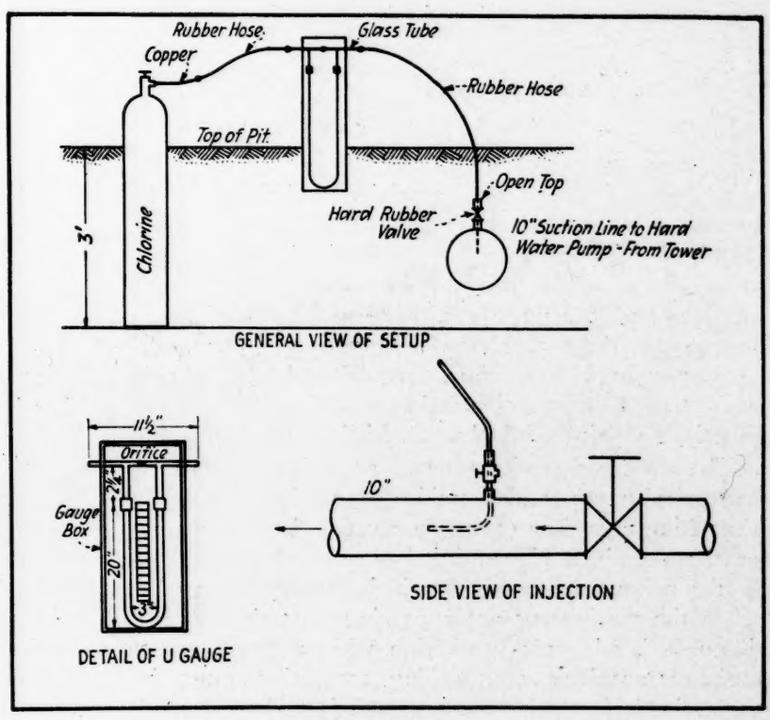
The means of injecting chlorine gas into a cooling tower system is very simple, as may be seen from the accompanying figure. It is simply metered through a glass orifice meter, constructed and calibrated in the laboratory, and then introduced into the water line leading to the circulating pump. This pump distributes the treated water uniformly over the top of the tower, thereby reaching and destroying the algae even in the most remote corners.

The particular system that we have been treating for the past year is at our Ventura compressor plant. The cooling tower is of the open type and contains about 50,000 gal. of water, circulating at 3000 g.p.m.

According to published data a concentration of 10 parts per million of chlorine in the water is sufficient to kill the algae.

When we first began our tests we used an orifice which permitted an injection of 2.5 lb. of chlorine per hour. The injection of this amount of chlorine into the water at the intake of the circulating pump enabled us to reach a concentration of from 5 to 10 p.p.m. in from two to five hours, depending upon the weather conditions and wind velocity. The cooling tower was badly infected with algae at the start of the tests, but by repeating these injections twice a week we found that the algae gradually decreased for about a 60-day period. From that time on the algae growth increased and appeared to be unaffected by altering the injection periods from daily to once a week. After several months operation the tower was thoroughly cleaned out and the chlorine injection was started again. In a short time the algae started to grow again and it was believed this was partly due to the fact that it being summer, the number of light hours per day was increasing. At times it was felt that the algae were again under control as a result of our altering the injection periods during the week, but after another five months the tower became so clogged it was necessary to shut it down and thoroughly clean it out once more.

After going over the results it was concluded that the chlorine was being injected too slowly and the algae were therefore building up a resistance to it. In addition it was felt that due to the slow rate of injection a




*Arrangement of equipment for injecting chlorine gas into cooling tower water.*

large amount of the chlorine was being dissipated to the atmosphere before the 10 p.p.m. was reached. This of course resulted in an excessive use of chlorine.

In February of this year the size of the injection orifice was changed so that the gas could be introduced at the rate of 12.5 lb. per hr. At such a rate it is possible to reach the concentration of ten parts per million in about 25 minutes, using only about 5 lb. of chlorine, compared to the 10 to 13 lb. per injection required with the smaller orifice.

Since February, 1937, the gas has been injected at the rate of 12.5 lb. per hr. but once each week, and the cooling tower has shown no indication of algae growth up to the present time.

From our experiences to date we have reached the following conclusions:

(1) The rate of injection should be sufficiently rapid to reach a 10 p.p.m. concentration in not over 30 minutes.

(2) By injecting at the rate of 12.5 lb. per hr. the cost is cut about in half, and this rate kills all growth sufficiently to enable the tower to operate a longer time before another injection is required.

(3) The cost of the above described treatment is about $5 per month. The upkeep cost of injecting, removing copper tubing, etc., is about $15 per year. Labor cost is not considered, as it is handled by the regular plant foreman.

(4) The use of a periodic concentration of 10 p.p.m. has not been sufficient to cause any corrosion to the pump parts or fittings in the water system. The tests show that all traces of chlorine have left the water system in about one hour after the injections are stopped.

During the coming summer experiments will be made to determine if an injection rate higher than 12.5 lb. per hr. would tend to reduce still further the cost of treatment and possibly require an injection but about every ten days instead of the present seven. If such tests prove successful they will be reported upon at a future date.

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# Air Conditions for Growing Mushrooms

THE whole matter of growing mushrooms has tended to be surrounded with mystery, because of the unusual conditions which favor their growth—complete absence of light, substitution of composted manure for soil, and location of the beds in dark, cool places, such as abandoned breweries, lime kilns, tunnels and caves. Moreover, mushroom growing is an art rather than a science. No colleges give instructions on the subject. The crop is grown under special conditions, such as lack of sunshine, so essential to most crops, and special nutrients other than soil are required for its growth. Very few books on the subject are available and the successful growers have gained their knowledge largely from their own experience.

In spite of these handicaps, the conditions for the growth of cultivated mushrooms have been carefully standardized. No other commercial food crop requires such careful control of temperature, humidity and ventilation, and every grower, whatever his equipment, has to face these problems. Fumigation of the mushroom houses with sulphur, formaldehyde, or cyanide gas is now a universal practice, according to the "Industrial Bulletin" of Arthur D. Little, Inc.

The more progressive growers are using instruments for temperature and humidity control. The white threadlike mycelium or spawn must be kept at a temperature of about 75F until it has penetrated completely throughout the mushroom bed. During the period when the mushrooms are appearing, a temperature of 50 to 60F, adequate ventilation without draft, and a moist atmosphere approaching saturation must be maintained if a satisfactory growth is to be expected. The spawn is supplied to the industry by organizations which specialize in growing it on sterile media and transferring it to suitable nutrients for sale. In these specialized spawn factories, control instruments are used for the maintenance of uniform temperatures, and, to keep out foreign spores which might contaminate their cultures, the air for ventilation is more carefully filtered than it is in the operating rooms of modern hospitals.

The mushroom, like other fungi, has no green coloring matter and can produce no starch. In order to grow, it must have available nitrogenous food and available cellulose. It is unlike plants and resembles animals in that it is a high protein source free from starch and during growth consumes oxygen and gives off carbon dioxide. This explains the necessity for adequate ventilation.

Mushrooms are readily grown and plentiful from October to April. During these months the wholesale price has averaged about 30c per pound for the fancy white variety. Lack of refrigeration is the real reason for the seasonal price fluctuation, and from May to September the effect of hot weather on the industry has been indicated by prices as high as $1 per pound.

Fungus pests, insects, lack of temperature and humidity control, scarcity of nutrient materials, a high degree of centralization, and great seasonal price variations are among the problems of the industry. The future should see expansion in output from the present annual crop valued at about $5,000,000, produced by about 500 establishments, and more decentralization with wider markets, particularly if synthetic nutrients can be developed. Fewer seasonal price fluctuations should occur as refrigeration costs are reduced and the conditioning of growing chambers becomes more widely adopted.

# INDUSTRIAL EXHAUST VENTILATION

## Resistance of Pipes and Elbows

By JOHN L. ALDEN

SUCCESSFUL design of exhaust piping is largely a question of correct estimation of resistances. The suction at the fan is the sum of the suction at the hood and the cumulative resistances of the piping, elbows, tapers, tees, and other units in series between the hood and the fan. Underestimated resistance means restricted flow and, possibly, inadequate dust control. Overestimated resistance means increased flow, excessive power, and the risk of an overloaded motor. Accurate evaluation of resistance is an essential preliminary to the inevitable compromise between aerodynamic efficiency on the one hand and low construction cost on the other.

The first step in the design of the system is the selection of hood types and dimensions from which entry characteristics can be estimated by the methods described in the preceding section. Branch pipe sizes and suction values are established next, either from experience or tests, from legal requirements or from Table 3. From the pipe size, hood suction, and entry coefficient can be calculated the air volume per minute entering each hood. For booths, canopy hoods, or other hoods requiring definite face velocities, the volume can be computed from the velocity and the face area. It must be understood that the volume rate of flow calculated at this stage is no more than an approximation based on hope or expectancy. It is an aid to judgment in arriving at a tentative arrangement of piping, fan, and dust separator. The next step is the preparation of the tentative layout in order to establish definite lengths and pipe diameters from which to calculate pressure drops and other factors affecting the final choice of dimensions and arrangement. To this stage, accuracy of calculation should be sacrificed to speed and labor saving.

### Conveying Velocity

The velocities maintained in branch pipes, mains. and discharge pipes should be low enough to minimize friction, yet high enough to prevent settlement of suspended solids. Table 5 lists a number of common materials handled by exhaust and low pressure conveying systems together with air velocities known to be adequate to transport them under ordinary circumstances. In a few instances the values have been taken from published data. The remainder are those in use in systems known to function satisfactorily.

The velocities quoted in Table 5 are, in general, safely high. They are not necessarily the most economical velocities. These can be obtained only by actual trial. In consequence, it is strongly recommended that the engineer check in his own plant the existing velocities in all systems handling similar materials in order to arrive at a still closer estimate of the most economical velocity. It is not enough to duplicate velocities already in use. The resourceful engineer can always devise simple means for temporarily reducing existing velocities to a level at which clogs begin to form and thereby arrive at minimum allowable values.

### Special Considerations

Finely divided organic solids usually constitute an explosion hazard. The same is true of powdered aluminum and some other metals which burn readily in the flaked or pulverized states. Systems transporting such materials, exemplified by flour, grain dust, wood flour, and phenolic molding powder, must be designed for velocities somewhat higher than the bare minima at which settlement occurs. Pockets formed by poorly proportioned enlargements must be avoided. Formation of eddies must be prevented so as to minimize stray deposits of material. These deposits may become explosion hazards or, in the case of poisonous dusts, health hazards to maintenance men.

Floor sweepings may introduce fire risk owing to the presence of tramp iron which may strike sparks from

TABLE 5—CONVEYING VELOCITIES FOR DUST COLLECTING

| Material | Velocity—F.P.M. | |
|---|---|---|
| | From | To |
| Wood flour—sander dust | 1500 | 2000 |
| Sawdust, light, dry | 2000 | 3000 |
| Sawdust, heavy, wet or green | 3000 | 4000 |
| Shavings, light, dry | 2000 | 3000 |
| Shavings, heavy, wet or green | 3000 | 4000 |
| Wood blocks, edgings, heavy, wet or green | 3500 | 4500 |
| Hog waste | 3500 | 4500 |
| Grinding dust | 3000 | 4000 |
| Foundry dust, tumbling barrel, shake-out | 3500 | 4500 |
| Sand blast dust | 3500 | 4000 |
| Buffing lint, dry | 2500 | 3000 |
| Buffing lint, sticky | 3000 | 4000 |
| Metal turnings | 4000 | 5000 |
| Lead dust | 4000 | 5000 |
| Cotton | 2500 | 3000 |
| Cotton lint, flyings | 1500 | 2000 |
| Wool | 3000 | 4000 |
| Jute lint, flyings | 2800 | 3000 |
| Jute picker stock, shredded bagging | 3000 | 3300 |
| Jute dust shaker waste | 3100 | 3400 |
| Jute butts (conveying) | 3600 | 4500 |
| Grain | 2500 | 4000 |
| Ground feed, ½-in. screen (conveying) | 4500 | 5000 |
| Grain dust | 2000 | 3000 |
| Coffee beans | 3000 | 3500 |
| Shoe dust | 3000 | 4000 |
| Rubber dust | 2000 | 2500 |
| Bakelite molding powder | 3000 | 3500 |
| Bakelite molding powder dust | 2000 | 2500 |

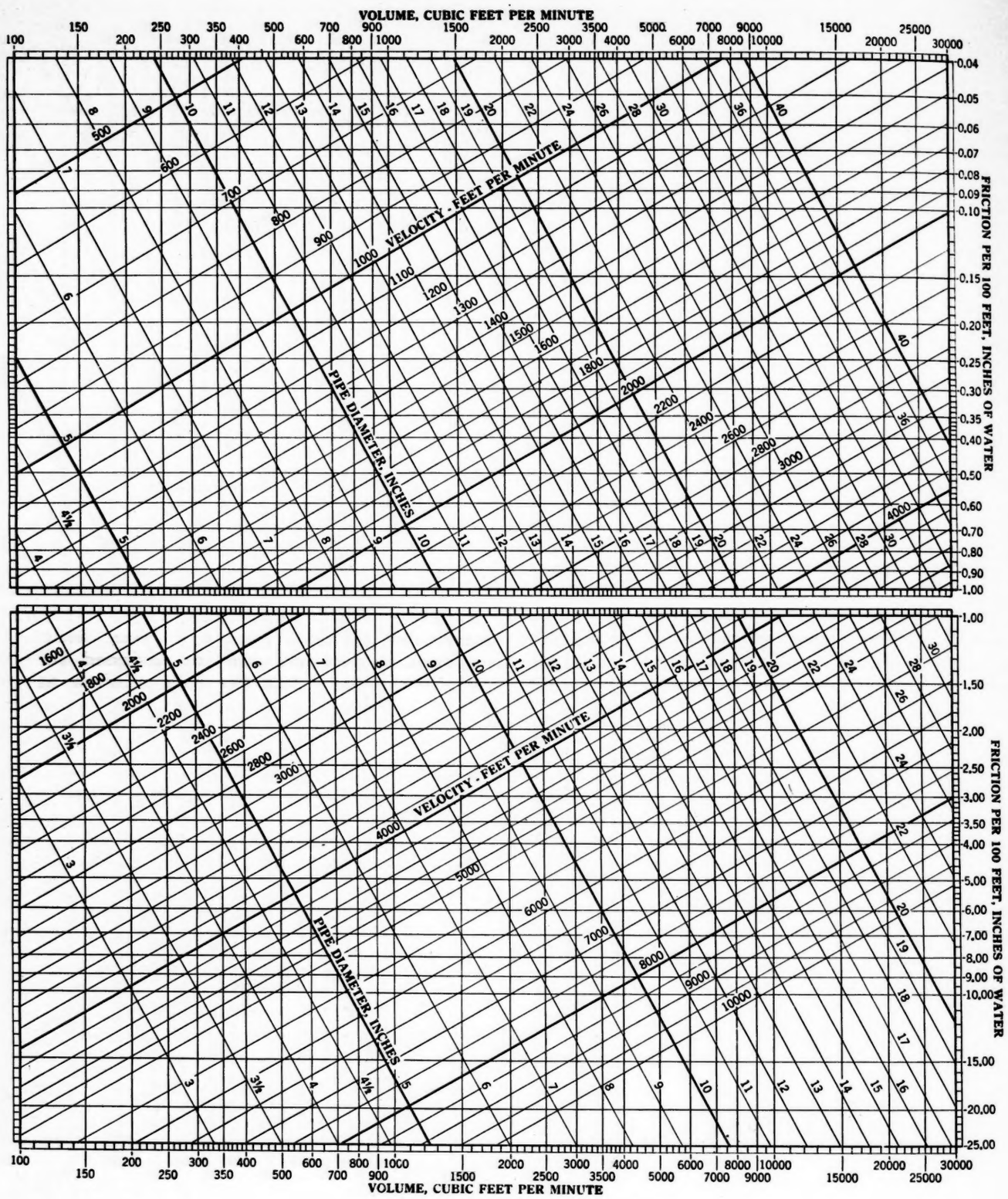

VOLUME, CUBIC FEET PER MINUTE
VELOCITY - FEET PER MINUTE
PIPE DIAMETER, INCHES
FRICTION PER 100 FEET, INCHES OF WATER
VOLUME, CUBIC FEET PER MINUTE

*Fig. 44. Friction loss in round pipes.*

the fan wheel or even from elbows. The designer of a system with floor sweep inlets will do well to limit velocities in floor sweep branches to those which will carry only the non-metallic constituents of the refuse. A further precaution is the provision of adequate traps between the inlet and the fan.

Cotton, lint, and hygroscopic substances drawn from warm humid rooms are troublesome to transport through outdoor piping during cold weather. Condensed moisture on the inner wall of the pipe causes the conveyed material to adhere to the cold, damp surface. Relief from clogs can be had only by drawing the air supply from out of doors or by increasing the velocity so as to reduce the time of exposure and to scour the pipe walls.

The practice of installing blast gates complicates the selection of conveying velocities. Closed gates disturb the balance of the system by reducing the velocity in portions of the mains and branches and increasing it in others. Consequently the transport velocity must be

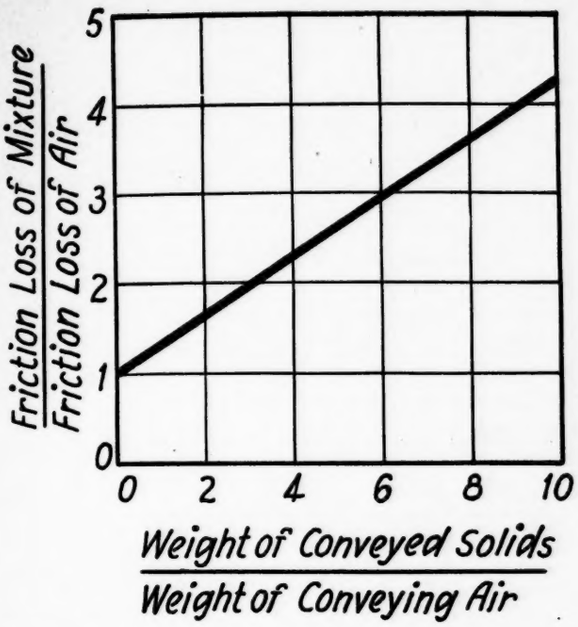



*Fig. 45. Friction loss of mixture expressed in terms of friction loss of clean air.*

sufficiently high in the most critical section to carry the material with any probable combination of closed gates.

### Friction Loss in Round Pipes

The dimensional formula for friction[1] may be reduced to the form

$$h_f = 0.0000747 \frac{fV^2}{D}$$

for standard air, where

$h_f$ = friction loss per 100 ft. of round pipe, in. wg.
$f$ = friction factor, Fig. 6
$V$ = average velocity in pipe, f.p.m.
$D$ = pipe diameter, in.

To facilitate practical determination of friction loss, Fig. 44 has been prepared. It will be observed that this chart relates four factors, volume rate of flow, velocity, pipe diameter, and friction loss per 100 ft. From any two factors the other two may be obtained. Thus, if a 15-in. pipe carries 4000 c.f.m., the velocity is close to 3300 f.p.m. and the friction loss per 100 ft. of pipe about 1.08 in. wg. The chart correctly reflects the dimensional formula within about 2% for densities corresponding to the temperatures between 32F and 100F. Uncertainties in the value of the friction factor, $f$, as well as the effects of design approximations and structural faults in the system are sufficient to obscure this error.

### Friction Loss in Rectangular Pipes

While rectangular pipes are used in exhaust systems much less frequently than round pipes, conditions occur which are best satisfied by this form. Friction losses in rectangular, annular, and oval ducts may be obtained by finding from Fig. 44 the loss in an aerodynamically equivalent round pipe. For shapes other than round, a factor, $m$, known as the "hydraulic radius," may be substituted for the pipe diameter in the dimensional formula for friction. The hydraulic radius is defined as the ratio of the cross sectional area to the perimeter of the duct. When $a$ and $b$ are the width and depth, respectively, of the duct, expressed in units consistent with the formula,

$$m = \frac{ab}{2(a+b)} \text{ for rectangular pipes and}$$

$$m = \frac{D}{4} \text{ for round pipes}$$

A rectangular pipe is frictionally equivalent to a round pipe when the velocities in both pipes are the same and when the hydraulic radii are equal. Hence, the diameter of a round pipe equivalent to a given rectangular duct is

$$D = \frac{2ab}{a+b}$$

when the friction losses per 100 ft. are equal. Example: Rectangular duct, 15 × 30 in., $Q$ = 10,000 c.f.m. Hence, $V$ = 3200 f.p.m. Find friction loss per 100 ft.

$$D = \frac{2 \times 15 \times 30}{15 + 30} = 20 \text{ in.}$$

From Fig. 44, friction loss per 100 ft. of 20-in. pipe at 3200 f.p.m. is 0.74 in. wg. This is also the required friction loss in the rectangular pipe.

### Friction Loss of Mixtures

It was shown in the section "Flow of Fluids"[2] that the presence of suspended solids in the ordinary dust collecting concentrations can have but slight effect on friction losses or power consumption. Concentrations commonly found in low pressure conveying systems, on the other hand, materially affect pipe friction. For example, the pneumatic elevating system of a common

[1] HEATING & VENTILATING, September, 1937, page 43.

[2] HEATING & VENTILATING, September, 1937, page 45.

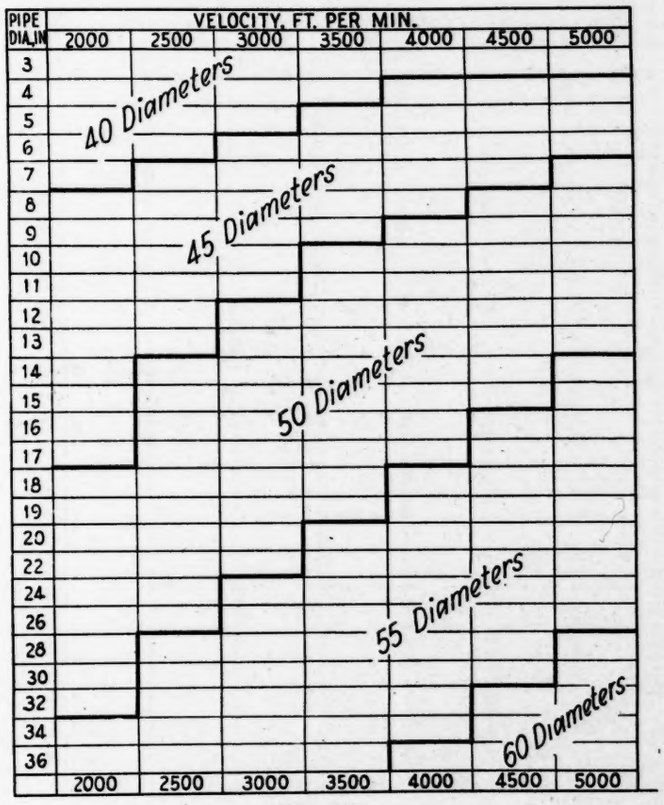



*Fig. 46.*

hammer mill handled 16,000 lb. of ground feed per hour through 5-in. diameter pipe with an air flow rate of 900 c.f.m. This is equal to 4 lb. of solids per pound of air. It was observed that the friction loss at this conveying rate was approximately 2½ times that when air alone was flowing.

The laws of flow of non-homogeneous fluids are not well established. Nevertheless, scattered practical tests provide at least an approximate working basis. Certain German experiments on conveying wheat[3] furnish the data from which Fig. 45 is constructed. The ratio of friction loss of the mixture to the friction of air alone holds good only for velocities high enough to carry the material completely in suspension. At lower velocities the ratio is greater, probably because the slip velocity of the air stream past the slowly moving solids dragging on the bottom of the pipe is high.

To obtain the approximate friction loss of a mixture, the loss when handling air only is read from Fig. 44. This then is multiplied by the appropriate factor from Fig. 45 corresponding to the expected weight of conveyed material per pound of conveying air.

### "Velocity Head Rule" for Pipe Friction

A rule often quoted for the resistance of straight round pipes is that the static pressure drop in a length of, say, 50 diameters, is equal numerically to one velocity head. While this rule has the virtue of simplicity, it must be used with discretion since it can be valid over a narrow range only. It has been shown in the section "Flow of Fluids"[4] that the friction factor, *f*, varies with the Reynolds number. It follows that the length of straight pipe, measured in diameters, to produce a loss of one velocity head also varies with the Reynolds number. Hence, no single value, such as 50 diameters, applies to the full range of pipe sizes and velocities common to exhaust systems.

If modified to use a sliding scale of equivalent lengths, the velocity head rule is useful in the calculation of expected flow through branch pipes. The diagram, Fig. 46, has been prepared from the same data as the friction loss chart, Fig. 44, and shows, for various velocities, the average equivalent length of pipe in diameters corresponding to the usual range of pipe sizes. Since the chief application of the rule is to branch pipes and since the customary velocity range lies between 3000 f.p.m. and 4000 f.p.m., the constants for branch pipe estimating can be memorized as follows:

TABLE 6—LENGTH OF BRANCH PIPE EQUIVALENT TO LOSS OF 1.0 VELOCITY HEAD

| Pipe Dia., In. | Equivalent Length, Diameters |
|---|---|
| Under 5 | 40 |
| 5 to 9 incl., | 45 |
| 10 and over | 50 |

### Friction Loss in Hose

No reliable friction loss data can be given for flexible hose. The internal roughness varies with the construction and condition. Moreover, hose is seldom used as straight pipe so that accurate prediction of loss is further complicated by bends. The cross-section at bends is likely to be flattened and of less than full area. The safe procedure is to purchase hose of smooth interior, bend it to the approximate form it will take in service and measure the actual pressure drop under at least one condition of flow. For other velocities the loss will be approximately in proportion to the velocities squared.

[3]"Experimental Investigation of a Pneumatic Grain-Conveying Unit," by Dr. Ing. Gasterstadt, Zeit., V.D.I., vol. 68, no. 24, June 14, 1924, page 617.

[4]Heating & Ventilating, September, 1937, page 43.

TABLE 7—RESISTANCE OF SQUARE AND ROUND 90° ELBOWS

| Center-Line Radius, Pipe Diameters | Elbow Loss, Velocity Heads | | | | Straight Pipe Equivalent To Elbow Followed By Duct, Pipe Diameters‡ | |
|---|---|---|---|---|---|---|
| | Elbow Followed By Duct† | | Elbow Discharging To Atmosphere (No Duct) | | | |
| | Round | Square | Round | Square | Round | Square |
| 0.0* | 0.95 | 1.15 | — | — | — | — |
| 0.5 | 0.83 | 1.05 | 1.54 | 1.95 | 37 | 47 |
| 0.625 | 0.55 | 0.58 | 1.18 | 1.25 | 25 | 26 |
| 0.75 | 0.41 | 0.37 | 0.96 | 0.87 | 18 | 17 |
| 1.0 | 0.28 | 0.21 | 0.72 | 0.54 | 13 | 9 |
| 1.5 | 0.24 | 0.13 | 0.65 | 0.35 | 11 | 6 |
| 2.0 | 0.21 | 0.11 | 0.55 | 0.29 | 9 | 5 |
| 3.0 | 0.21 | 0.11 | 0.42 | 0.22 | 9 | 5 |

†Elbow loss only. Does not include duct loss.
‡Based on loss of 1.0 velocity head in 45 diameters of straight pipe.
*Mitre or "stove pipe" elbow.

### Losses in Elbows

Losses in exhaust fittings such as elbows, tees, and tapers can be estimated less accurately than can resistance of straight pipe. Experimental information is meager and much of that available is of questionable reliability. In the absence of more complete data, the following material provides a reasonable estimating basis. The quoted losses are expressed as functions of the velocity head although it is believed that they vary with the Reynolds number in much the same way as do pipe resistance and orifice characteristics.

Losses in round and rectangular elbows as determined by Busey[5] have been used successfully for many years. More recent work by Madison and Parker[6] indicates that Busey's values for rectangular elbows are somewhat low. Independent measurements on round elbows suggest that the figures for these also should be increased. Consequently, Table 7 quotes the Madison and Parker values for rectangular elbows and figures proportionally higher than Busey's for round elbows.

TABLE 8—EFFECT OF ASPECT RATIO ON ELBOW RESISTANCE

| Aspect Ratio | Relative Resistance | |
|---|---|---|
| | Rad. Ratio = 0 to 0.5 | Rad. Ratio = 1.0 to 3.0 |
| 0.25 | 1.30 | 2.15 |
| 0.5 | 1.15 | 1.35 |
| 1.0 | 1.0 | 1.0 |
| 2.0 | 0.90 | 1.0 |
| 3.0 | 0.80 | 0.95 |
| 4.0 | 0.75 | 0.90 |

[5]"Loss of Pressure Due to Elbows in the Transmission of Air Through Pipes and Ducts," by F. L. Busey, ASHVE Transactions, vol. 19, 1913, page 366.

[6]"Pressure Losses in Rectangular Elbows," by R. D. Madison and J. R. Parker, ASME Transactions, vol. 58, no. 3, April, 1936, pages 167-176.

The losses are expressed in terms of equivalent length of straight pipe as well as in velocity heads.

The degree of "hardness" of bend of an elbow is usually designated by expressing the centerline radius or throat radius in terms of pipe diameter. A "long-radius" elbow may be considered as one whose centerline radius is 2.0 diameters or greater. The terms "radius ratio" and "curve ratio," illustrated in Fig. 47, are less often used to designate the character of the bend.

The velocity of air flowing along the outside of a bend is not only higher than that of the filaments nearer the throat but the stream, in addition, shows a pronounced tendency toward a double helical motion symmetrical about the plane of the bend. The plaster flow casts prepared by Wirt[7] strikingly illustrate the nature of elbow flow. When the elbow is followed by a short length of straight pipe, there is a partial recovery of static pressure as the velocity diminishes and the helical flow subsides. When the elbow discharges directly to the atmosphere, no static pressure regain takes place. Table 7 shows that the loss when followed by 4.0 diameters of pipe is less than that when discharging to the atmosphere. The loss quoted for the elbow followed by a duct is for the elbow only and does not include the duct loss.

### Influence of Aspect Ratio

The aspect ratio is a term expressing the width of an elbow cross-section in terms of its depth as shown in Fig. 47. It applies to rectangular elbows only. Square elbows have an aspect ratio of 1.0. "Easy" turns have aspect ratios greater than 1.0. Table 8 shows that high aspect ratios are accompanied by low elbow losses.

### Influence of Angle of Turn

Table 7 refers only to 90° elbows. In Table 9 is shown the variation of resistance with the angle of bend. The data have been obtained for rectangular elbows but are approximately correct for round elbows also. It will be observed that the losses in bends greater than 90° are less than would be expected from direct proportionality.

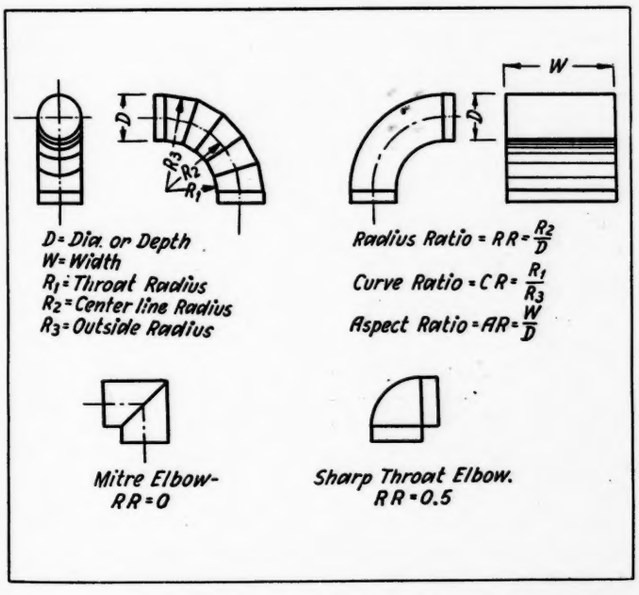
[7] "New Data for the Design of Elbows in Duct Systems," by Loring Wirt, General Electric Review, vol. 30, June, 1927, pages 286-296.

**TABLE 9—EFFECT OF ANGLE OF BEND ON ELBOW RESISTANCE**

| Angle of Bend, Degrees | Relative Resistance | |
|---|---|---|
| | Elbow Followed By Duct | Elbow Discharging To Atmosphere |
| 15 | 0.18 | 0.18 |
| 30 | 0.34 | 0.35 |
| 45 | 0.52 | 0.53 |
| 60 | 0.68 | 0.69 |
| 75 | 0.84 | 0.86 |
| 90 | 1.00 | 1.00 |
| 120 | 1.27 | 1.22 |
| 150 | 1.50 | 1.34 |
| 180 | 1.65 | 1.42 |

### Influence of Approach Velocity

If the velocity distribution in the pipe approaching the elbow departs materially from the nearly parabolic profile shown in Fig. 8, the elbow loss may differ considerably from the tabulated values. Elbow losses are particularly sensitive to flow disturbances in the region of the elbow throat. High approach velocities along the inner radius tend to increase the loss whereas high velocities along the outer arc decrease it. In consequence, the arrangement of elbows directly connected to fan outlets has considerable bearing on the elbow losses. Air flows through the wheel and casing of a centrifugal exhauster in such a way as to produce in the fan outlet a distorted velocity pattern. Exit velocities are highest along the scroll sheet of the casing and in the plane of the wheel back plate. Hence, when a discharge elbow is so oriented that its throat receives the high velocity flow from these regions, excessive loss may be expected. Thus, high losses, as much as twice normal, have been found when the elbows were arranged as in Fig. 48a and c. Losses less than normal may be expected from arrangements such as Fig. 48b and d.



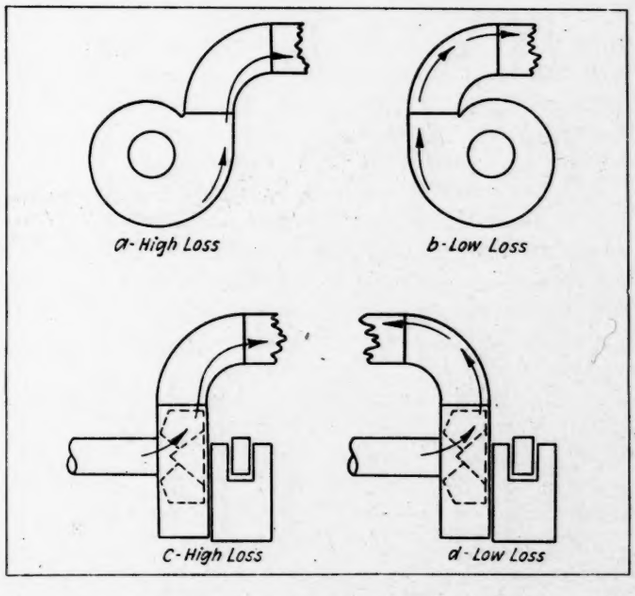




*Fig. 47 (Left). Radius-, curve-, and aspect ratios of round and rectangular elbows. Fig. 48 (Right). Effect of fan discharge arrangement on elbow loss.*

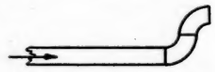
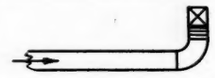
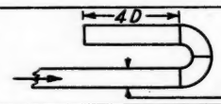
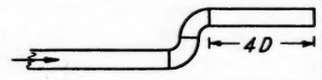
| Symbol | Type of Compound Elbow | Ratio Measured to Predicted Loss | |
|---|---|---|---|
| | | Elbows Direct Connected as Shown | Elbows Connected by Pipe 1 Dia Long |
| a | | 1.04 | 0.55 |
| b |  | 0.94 | 1.06 |
| c |  | 0.66 | 0.75 |
| d |   | 0.83 | 0.60 |
| e |   | 1.20 | 1.31 |
| f | | 0.81 | 0.88 |

*Fig. 49 (Left). Losses in compound elbows relative to the sums of expected losses in single elbows.*

### Compound Elbows

Two elbows joined together in tandem or connected by a short length of pipe usually impose a loss for the combination appreciably different from the sum of the expected losses of the individual elbows as quoted in Table 7. Madison and Parker tested a number of combinations some of which are shown in Fig. 49. In comparing measured and predicted losses, the loss of the upstream elbow has been assumed to be that of an elbow followed by a duct. In Fig. 49a, b, and c the estimated loss of the downstream elbow is that of an elbow only. In Fig. 49d, e, and f the estimated loss is that of an elbow followed by four diameters of ducts.

The usefulness of Fig. 49 in exhaust work is qualitative rather than quantitative. The quoted tests were made on elbows of square cross section and of relatively short centerline radii. The losses of long radius round elbows in combination may be materially different. Nevertheless, in the absence of better information, Fig. 49 may be used as a guide to the losses of compound elbows.

## Effect of Chemical Treatments on the Burning of Coal

CHEMICAL mixtures designed for the treatment of coal, with a view to making coal burn better, really have little effect on the combustion of the fuel, it has been found by engineers of the Bureau of Mines, Department of Interior, after making exhaustive tests.

Patent records show that the idea that the burning of fuels can be improved by mixing or pretreating them with chemicals dates back to the early years of the last century. In the earlier years the amount of chemical recommended to be used was large, but from 1860 on it has ranged from ½ to 4 lb. per ton of coal. Until recent years the sale of such products was restricted to packages sold to householders under the general name of "fuel savers" with the statement that they would reduce the amount of coal used from 10 to 40%; during the last 10 years their use has extended to treating coal for large boilers.

The Bureau of Mines has continually received inquiries about the efficacy of these products, and in response to these requests, the Bureau took up this subject of investigation and thoroughly tested, both alone and in mixtures, all the chemicals known to have been marketed for this purpose, and many others, including water and chlorine. The chemicals were applied to a wide range of bituminous coals and to high-temperature and low-temperature cokes. The untreated and treated fuels were burned, under carefully controlled conditions, in overfeed and underfeed fuel beds at rates of burning varying from 1 to 50 lb. of fuel per square foot of grate per hour.

Factors studied in detail were the effects of the chemicals on rates of ignition and combustion, on caking of coal in fuel beds, on tar and soot carried from the fuel beds, on soot deposited on surfaces of the furnaces and flues, on the emission of smoke, on the ash and clinker, and on the reaction of the sulphur in the fuels.

The tests of this investigation and the deductions from them show that the possible fields of application of the pretreatment of fuels with any chemical are extremely limited. The effects of light treatments of the order of 4 lb. of chemical alone per ton of fuel, apart from the water that might accompany its use, were so small under any conditions of burning that they could not account for the improvement that users of treatments in this or smaller amounts have believed they observed in service. The effects are about proportional to the amount of treatment applied. No efficacy was found for a mixture of chemicals, and each ingredient seemed to act in proportion to the amount that was present. Too much chemical applied to the surfaces of a fuel smothered it and hindered ignition and burning. The tests also showed that any changes that treatments can make in the burning of coal or coke decrease in magnitude as the rate of burning increases, and that differences are small at rates of burning higher than 10 to 15 lb. of coal per square foot of grate per hour, even with treatments as high as 40 lb. per ton of fuel.

These findings do not in any way relate to the effectiveness of any material in reducing coal dustiness.

The results of these investigations are given in Bulletin 404, "Burning of Coal and Coke Treated with Small Quantities of Chemicals," by P. Nicholls, W. E. Rice, B. A. Landry, and W. T. Reid, copies of which may be obtained from the Superintendent of Documents, Government Printing Office, Washington, D. C., at a price of 15 cents each.

# Lowest Temperatures of Record

The accompanying map and table are devoted to data on the lowest temperatures ever recorded at official weather stations. The table covers 100 of the largest cities, while the map is based on readings from a very large number of stations and is a simplification of an official Weather Bureau map. The map is believed to be substantially correct through 1933.

***Wind Also Should be Considered***

The lowest temperatures on record are of concern to the heating engineer in designing plants, but it should be noted that the lowest temperature on record does not constitute the whole picture of severity of load. A higher temperature than the lowest ever recorded but with a sharp wind may give a greater load than when the temperature is extremely low and the air is still.

Table 2 is a study of weather conditions during the season of 1933-34 when a number of low temperature records were broken in the east and middle west. The table shows the lowest temperatures attained that winter and the wind conditions when the low temperature was reached. The last column shows the average (over a long period) wind velocity for December, January, and February. Study of Table 2 shows that the wind velocity at the time of coldest temperature was actually greater than the winter average for 10 of the 22 cases. In other words, contrary to common belief, higher-than-average wind velocities often accompany low temperatures. Of these 22 cases, the wind in five cases was from an entirely different direction than the prevailing winter wind.

***Low Temperatures by States***

Table 1 shows the lowest temperatures on record by states covering the period up to and including the year 1933. The lowest temperature ever reported in the United States was —66F at the Riverside Ranger Station in Yellowstone Park, Wyo., on February 9, 1933.

TABLE 1—LOWEST TEMPERATURES EVER REPORTED, BY STATES

| | | | | | |
|---|---|---|---|---|---|
| Alabama | —18 | Maine | —48 | Ohio | —39 |
| Arizona | —33 | Maryland | —40 | Oklahoma | —27 |
| Arkansas | —29 | Massachusetts | —26 | Oregon | —54 |
| California | —36 | Michigan | —49 | Pennsylvania | —42 |
| Colorado | —54 | Minnesota | —59 | Rhode Island | —18 |
| Connecticut | —25 | Mississippi | —16 | South Carolina | —11 |
| Delaware | —17 | Missouri | —40 | South Dakota | —57 |
| Florida | —2 | Montana | —65 | Tennessee | —32 |
| Georgia | —12 | Nebraska | —47 | Texas | —23 |
| Idaho | —57 | Nevada | —45 | Utah | —50 |
| Illinois | —35 | New Hampshire | —46 | Vermont | —44 |
| Indiana | —33 | New Jersey | —34 | Virginia | —29 |
| Iowa | —47 | New Mexico | —48 | Washington | —40 |
| Kansas | —40 | New York | —46 | West Virginia | —37 |
| Kentucky | —30 | North Carolina | —21 | Wisconsin | —54 |
| Louisiana | —16 | North Dakota | —56 | Wyoming | —66 |

# LOWEST TEMPERATURES EVER RECORDED

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# LOWEST TEMPERATURE EVER RECORDED IN 100 CITIES

| City | Lowest Temp., F |
|---|---|
| Albany, New York | −24 |
| Albuquerque, New Mexico | −10 |
| Atlanta, Georgia | −8 |
| Baltimore, Maryland | −7 |
| Birmingham, Alabama | −10 |
| Boise, Idaho | −28 |
| Boston, Massachusetts | −18 |
| Buffalo, New York | −14 |
| Burlington, Vermont | −29 |
| Butte, Montana | −52 |
| Charleston, South Carolina | 7 |
| Charlotte, North Carolina | −5 |
| Cheyenne, Wyoming | −38 |
| Chicago, Illinois | −23 |
| Cincinnati, Ohio | −17 |
| Cleveland, Ohio | −17 |
| Columbia, South Carolina | −2 |
| Columbus, Ohio | −20 |
| Concord, New Hampshire | −35 |
| Dallas, Texas | −3 |
| Denver, Colorado | −29 |
| Des Moines, Iowa | −30 |
| Detroit, Michigan | −24 |
| Duluth, Minnesota | −41 |
| El Paso, Texas | −5 |
| Erie, Pennsylvania | −15 |
| Eugene, Oregon | −4 |
| Evansville, Indiana | −16 |
| Fitchburg, Massachusetts | −17 |
| Flagstaff, Arizona | −25 |
| Fort Smith, Arkansas | −15 |
| Fort Wayne, Indiana | −24 |
| Frederick, Maryland | −21 |
| Grand Forks, North Dakota | −43 |
| Grand Rapids, Michigan | −24 |
| Great Falls, Montana | −49 |
| Greensboro, North Carolina | −3 |
| Hartford, Connecticut | −18 |
| Houston, Texas | 5 |
| Huntington, West Virginia | −24 |
| Indianapolis, Indiana | −25 |
| Jacksonville, Florida | 10 |
| Kansas City, Missouri | −22 |
| Knoxville, Tennessee | −16 |
| Lexington, Kentucky | −20 |
| Lincoln, Nebraska | −29 |
| Little Rock, Arkansas | −12 |
| Los Angeles, California | 28 |
| Louisville, Kentucky | −20 |
| Madison, Wisconsin | −29 |
| Memphis, Tennessee | −9 |
| Meridian, Mississippi | −6 |
| Milwaukee, Wisconsin | −25 |
| Minneapolis, Minnesota | −34 |
| Minot, North Dakota | −49 |
| Mobile, Alabama | −1 |
| Newark, New Jersey | −13 |
| New Orleans, Louisiana | 7 |
| New York, New York | −13 |
| Norfolk, Virginia | 2 |
| Oklahoma City, Oklahoma | −17 |
| Omaha, Nebraska | −32 |
| Paterson, New Jersey | −13 |
| Pensacola, Florida | 7 |
| Peoria, Illinois | −27 |
| Philadelphia, Pennsylvania | −11 |
| Phoenix, Arizona | 16 |
| Pittsburgh, Pennsylvania | −20 |
| Portland, Maine | −21 |
| Portland, Oregon | −2 |
| Providence, Rhode Island | −17 |
| Pueblo, Colorado | −27 |
| Reading, Pennsylvania | −14 |
| Reno, Nevada | −19 |
| Richmond, Virginia | −3 |
| Rochester, New York | −14 |
| St. Louis, Missouri | −22 |
| Salt Lake City, Utah | −20 |
| San Antonio, Texas | 4 |
| San Diego, California | 25 |
| San Francisco, California | 27 |
| Savannah, Georgia | 8 |
| Scranton, Pennsylvania | −19 |
| Seattle, Washington | 3 |
| Shreveport, Louisiana | −5 |
| Sioux City, Iowa | −35 |
| Sioux Falls, South Dakota | −42 |
| South Bend, Indiana | −20 |
| Spokane, Washington | −30 |
| Springfield, Illinois | −24 |
| Syracuse, New York | −22 |
| Topeka, Kansas | −25 |
| Trenton, New Jersey | −13 |
| Tulsa, Oklahoma | −16 |
| Vicksburg, Mississippi | −1 |
| Washington, District of Columbia | −15 |
| Wheeling, West Virginia | −18 |
| Wichita, Kansas | −22 |
| Wilmington, Delaware | −15 |
| Yuma, Arizona | 22 |

## TABLE 2—TEMPERATURE AND WIND CONDITIONS AT TIME OF LOW TEMPERATURE

| City | TEMPERATURE AND WIND CONDITIONS, 1933-34 | | | NORMAL WINTER WIND |
|---|---|---|---|---|
| | 1933-34 Lowest Temp., F | 1933-34 Date of Lowest Temp. | Wind Vel. (m.p.h.) and Direction at Time of Low Temp. | Av. Vel. and Direction |
| Birmingham, Ala. | 10 | Jan. 30 | 7 N | 8.6 N |
| Ft. Smith, Ark. | 15 | Feb. 27 | 12 E | 8.0 E |
| Los Angeles, Calif. | 40 | Dec. 25 | 4 NW | 6.3 NE |
| San Francisco, Calif. | 39 | Dec. 25 | 11-17 N | 7.6 N |
| Hartford, Conn. | — 15 | Feb. 9 | 7 NW | 8.5 NW |
| Washington, D. C. | — 7 | Feb. 9 | 9 N | 7.1 NW |
| Atlanta, Ga. | 7 | Jan. 30 | 17 NW | 12.0 NW |
| Chicago, Ill. | — 9 | Dec. 27 | 7-9 W | 12.5 W |
| Dubuque, Iowa | — 18 | Dec. 27 | 4 S | 7.1 NW |
| Portland, Me. | — 18 | Dec. 29 | 21 N | 9.2 NW |
| Baltimore, Md. | — 7 | Feb. 9 | 21 NE | 7.8 NW |
| Boston, Mass. | — 18 | Feb. 9 | 21 NW | 11.2 W |
| Detroit, Mich. | — 16 | Feb. 9 | 5 NE | 12.7 SW |
| Duluth, Minn. | — 32 | Dec. 25 | 11 W | 12.6 SW |
| St. Louis, Mo. | 2 | Feb. 27 | 5 W | 11.6 S |
| Atlantic City, N. J. | — 9 | Feb. 9 | 24 NW | 16.0 NW |
| New York, N. Y. | — 14 | Feb. 9 | 15 N | 17.1 NW |
| Cleveland, Ohio | — 11 | Feb. 9 | 9 NE | 13.0 SW |
| Philadelphia, Pa. | — 11 | Feb. 9 | 24 N | 11.0 NW |
| Providence, R. I. | — 17 | Feb. 9 | 15 NW | 12.8 NW |
| Burlington, Vt. | — 30 | Dec. 30 | 6 E | 11.5 S |
| Seattle, Wash. | 32 | Jan. 8 | 5 SE | 11.3 SE |

# Moisture Difficulties in Modern Construction

By E. R. QUEER†

THE public is becoming heat insulation and air conditioning conscious. It appreciates the economic value as well as the comfort obtained from both of these relatively new innovations. Insulation has come to be recognized as one of the essentials of good modern construction. Evidence of its wide acceptance is shown in statistics which indicate that a much greater portion of the construction dollar is now being spent on this item than during the building boom years, 1926-1928. Along with insulation, air conditioning, particularly for winter use, is becoming popular in dwellings.

*Windowless building housing the Thermal Testing Laboratory at the Pennsylvania State College.*

Humidification, an essential part of winter air conditioning, is raising some problems concerning heat insulation and structural parts of buildings, which must be given serious consideration. Winter climatic conditions in the northern part of this country and humidification within buildings combine to set up unremitting forces which condense water on windows and may deposit moisture within the walls of structures. Condensation on windows is a nuisance and will finally lead to destruction of paint, rotting out of wood sash, or the corrosion of metal sash. The deposition of moisture in walls will not only destroy the economic value of insulation but may eventually cause serious deterioration of the structural members within walls. If we hope to maintain a fair degree of permanence with modern building, it is necessary to give careful consideration to construction methods for alleviating these conditions.

### Condensation and Frost on Windows

Considerable progress has been made with the forced air heating system and it is receiving widespread attention in residence construction. It so happens that this system is perhaps more adaptable to both winter and summer air conditioning than are the others. Consequently, the majority of new residences have all the essentials of winter air conditioning. Since winter climatic conditions in most of the northern half of this country are such as to make it necessary to supply humidification and heat to air to produce comfort, we are faced with troublesome condensation and frost on windows during cold weather.

For comfort within a dwelling during the winter current practice considers that the temperature should be 70 to 72F and the relative humidity between 45% as a practical maximum, and 35% the recommended minimum. When the outdoor temperature is 37F and the relative humidity 45%, condensation forms on the inside of single windows. Condensation appears on a single window at 26F outdoors and 35% relative humidity within. Whereas, with double glazing or winter windows, condensate should not form until the outdoor temperature drops to –10F with 45% relative humidity inside. When the relative humidity is 35% the outdoor temperature can drop as low as –40F before trouble is experienced with condensation. Buildings fitted with single windows must have the relative humidity reduced to 18% in zero weather to prevent condensation. Inasmuch as humidified air is an essential part of winter air conditioning, it is evident that to take full advantage of humidification, double glazing or winter windows must be installed.

Winter windows have the added advantages of increasing comfort and effecting material savings in fuel costs, particularly when expensive fuels are burned. If one sits close to an undraped single window during subzero weather, he feels chilly in spite of 75F surrounding air, whereas, a winter window will raise the inner glass temperature sufficiently to produce a feeling of comfort. Winter windows not only reduce direct heat losses from a structure but when effectively installed, infiltration losses are also reduced. About one-half of the total heat loss from single window buildings is chargeable to the windows and doors, in the form of direct heat loss and infiltration. Since this is the case, it is useless to over-insulate the walls and roofs of buildings without doing something to improve the windows and doors. A balanced condition of insulation should be distributed over all the exposed surface of the building. A single window will lose about five times as much heat as a like area of ordinary uninsulated wall construction; while double windows lose only about twice as much. The substantial savings in fuel cost thus provided, when expensive fuel is burned, make possible a quick amortization of the investment in winter windows.

If condensation and frosting are permitted to continually take place on windows during cold weather, much damage may be expected to result. The paint on frames and the stools may peel

**Insulation of side walls and ceilings is commonly practiced, but the author believes that more effective heat insulation at the windows should be considered. He also shows by calculated analysis the nature of the problem of moisture in insulation and suggests methods of eliminating it.**

†Engineering Experiment Station, The Pennsylvania State College.

off and in the case of steel frames rust soon appears. Cases are known where wooden frames and stools deteriorated to such an extent from rotting that they had to be replaced within ten years after installation. Furthermore, the water condensed on the windows sometimes drains down into the walls beneath the window and causes disintegration of structural members as well as discoloring of the plaster. Since it is not desirable to reduce the humidity within the building, winter windows should be installed to raise the inside glass temperature above the dewpoint and eliminate condensation.

Frequently, winter windows are installed only on the windward side of a building and the leeward side is neglected. This undoubtedly accounts for the name "storm window" having been adopted. It should be remembered that when the wind blows, there is just as much negative pressure created on the leeward side of a structure as there is positive pressure built up on the windward side. Therefore, heated air is sucked out of the building to the same extent as cold air is forced in on the windward side. It should be kept in mind that condensation also forms on leeward windows when the outdoor temperature is low enough. In order to obtain the maximum saving in heat and comfort, it is obvious that winter windows or double glazing should be installed on all windows.

Winter windows are installed either on the outside of the permanent window or inside the permanent window. Both ways should produce the same reduction in heat losses; however, in the latter case, care must be taken to obtain a good vapor seal between the windows. When the seal is not sufficiently tight, vapor is drawn into the space by breathing action and condensed on the outer window. In one instance, this type of installation caused continual trouble to be experienced all winter long. Vision through the windows was obstructed by condensate and frost. Little difficulty is encountered when the winter window is placed outside the permanent window by the usual methods. Usually the permanent window is tight enough to prevent much breathing and such vapor as may be drawn into the space passes on to the outdoors because the winter window is not sealed tightly enough to trap the moisture.

### Moisture Accumulation in Walls

Since winter air conditioning has become so popular, we are faced with the problem of attempting to maintain dryness within outer walls of those buildings in which it has been installed. It is well known that ordinary building materials such as plaster, plaster bases, and most insulating materials when assembled in the walls of a building, do not form a vapor-impervious barrier. With humidified and heated air within and low outdoor temperatures, vapor will flow through the relatively porous walls from the region of high vapor pressure to that of lower vapor pressure outside. In passing through the walls it will encounter dewpoint temperatures and water will be condensed in these areas. Moisture may continue to accumulate and eventually saturation of the wall may occur. It is not difficult to visualize what the net effect of such a condition might be in a structure after a number of heating seasons. The insulating value of insulation would be destroyed and wood structural members would decay. The ultimate result would be a low degree of permanence of such buildings.

Cases are known where condensation within walls has produced water in sufficient quantities to run down over the sill plates into the basement. These same sill plates are, in the short space of three years since installation, showing signs of deterioration from rotting.

If the heat losses and temperatures within a few ordinary wall structures are analyzed, some enlightening facts are obtained regarding the conditions which may exist. The results of such an analysis made by calculations based on usual heat transfer values are shown in Table 1. The degree with which these results are reproducible in practice is dependent on the agreement between heat transfer values obtained in the laboratory on a dry basis and the values for the materials as installed. The quality of workmanship in erection will

TABLE 1—TEMPERATURES WITHIN WALLS

TEMPERATURES—INDOOR 70F, OUTDOOR 0F.
RELATIVE HUMIDITY 45%, DEWPOINT 52F.

| WALL NO. | AMOUNT AND TYPE OF INSULATION | LOCATION OF INSULATION | U | HEAT LOSS B.T.U. PER HR. SQ. FT. | TEMPERATURE | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | SURFACE PLASTER | UNDER PLASTER | AIR SPACE | | COLD SIDE SHEATHING | OUTSIDE |
| | | | | | | | HOT SIDE | COLD SIDE | | |
| 1 ..... | None | — | 0.243 | 17.0 | 59.5 | 57.0 | 52.4 | 36.9 | 19.9 | 2.9 |
| 2 ..... | ½ in. board | Plaster base | 0.188 | 13.2 | 61.9 | 59.9 | 40.5 | 28.6 | 15.4 | 2.2 |
| 3 ..... | 3⅝ in. fill | Stud space filled | 0.065 | 4.6 | 66.8 | 65.6 | 64.9 | 9.9 | 5.3 | 0.8 |
| 4* ..... | 1 in. blanket | Between studs, 2 air spaces | 0.126 | 8.8 | 65.0 | 63.6 | 61.3 | 19.2 | 10.3 | 1.5 |
| 5 ..... | ½ in. board | Sheathing | 0.216 | 15.1 | 60.8 | 58.5 | 54.4 | 40.7 | 17.7 | 2.6 |
| 6 ..... | 1 in. board | Sheathing | 0.165 | 11.6 | 62.9 | 61.2 | 58.1 | 47.6 | 13.6 | 2.0 |
| 7 ..... | Two ½ in. boards | Sheathing and plaster base | 0.173 | 12.1 | 62.6 | 60.8 | 43.0 | 32.0 | 14.2 | 2.1 |
| 8 ..... | ½ in. board 1 in. board | Plaster base sheathing | 0.138 | 9.7 | 64.3 | 62.8 | 48.6 | 39.7 | 11.3 | 1.6 |
| 9 ..... | 1 Alum. foil surface | Facing air space on plaster base side | 0.186 | 13.0 | 61.0 | 60.1 | 56.6 | 28.3 | 15.2 | 2.2 |
| 10 ..... | 1 Alum. foil surface ¾ in. board | Facing air space on plaster base side. Moisture-proofed sheathing | 0.152 | 10.6 | 63.6 | 62.0 | 59.1 | 35.9 | 12.4 | 1.8 |

*Temp. hot side blanket 53.2F; Temp. cold side blanket 27.2F.

also have a bearing on the agreement. As a standard wall, a common frame construction was chosen and it consists of siding, a good grade of building paper, sheathing, 2 x 4 studs, gypsum plaster base and plaster. The other walls have been varied from the standard by the addition and substitution of insulation for certain materials. In the occupied space of the building, a temperature of 70F and a relative humidity of 45% are assumed, and the outdoor temperature is chosen as 0F. Under these conditions moisture will condense on surfaces which are at 52F or lower temperatures. If the outside surface of the outer wall is sufficiently tight the humidity condition within the wall will approach that in the occupied quarters and moisture will be condensed in the wall. As an example, in Wall 1 (see Table 1), moisture would appear on the sheathing within the air space. In Wall 2, the temperature gradient through the wall is such that condensation should take place in the plaster base. In the majority of these typical wall constructions, the condensation will take place on the colder surface bounding the stud space. Since it is customary to carefully seal all air spaces within walls for fire protection, eventual saturation of structural members should be expected. These conditions are favorable for the promotion of rot and decay in organic materials and will in time lead to failure of members. Specific examples of such failures in relatively new construction are now being produced.

An additional problem along this same line arises in Wall 3. Here the space between the studs is filled with a batt or fill type of insulation. The temperature gradient in this wall places the condensation area within the insulation. The water will migrate in the direction of heat flow and will accumulate on the cold surface bounding the insulation. If the temperature is low enough, it will freeze on the surface.

Since humidification and insulation are both essential factors in modern buildings, methods must be found to overcome the difficulties involved. If an effective vapor barrier is placed on the plaster side of the wall the moisture cannot get into the wall. A reasonable amount of insulation should be used and the wall should be provided with an air space. A completely filled or solid wall does not permit air circulation within the structure and probably will not give the moisture a chance to be released once it is deposited. Although it is easier to completely fill stud spaces, it is not essential to use more than 2½ in. of insulation in most structures. It is also suggested that the outer wall structure be made less vapor-tight and that the stud space be vented to a region of lower vapor pressure. However, care must be observed in this procedure, otherwise the walls may be saturated from hard driving rains, and regulations regarding fire stops may be violated. Sometimes a combination of insulating materials such as indicated in Wall 10, Table 1, along with venting will avoid moisture accumulation. In this wall the hot side temperature of the air space is above the dewpoint and venting to a region of lower vapor pressure should reduce the moisture content of the air within the space.

### Conclusions

There is a tendency to use more and more humidification in living quarters during the winter heating season. Therefore it is necessary to install winter windows in order to prevent condensation and frosting of single windows. Manufacturers of winter air conditioning systems should recognize this fact and insist that winter windows be installed in conjunction with their equipment. Otherwise, it will not be possible to take full advantage of the equipment during the colder parts of the heating season when it is most needed. Furthermore, since about 50% of the heat losses from an ordinary house are directly chargeable to windows and doors, it is economically unsound to over-insulate the walls of a building and neglect the windows and doors. Winter windows not only reduce the direct heat loss through windows, but materially reduce infiltration when properly installed. It will also be found that the savings in expensive fuel costs gained from their use will amortize the investment in them in a few years.

Since winter air conditioning has come into prominence, condensation difficulties within the walls of relatively new buildings have been discovered. In future construction, these facts should be considered and efforts made to overcome them so that a fair degree of structural permanence may be expected. A good vapor seal should be placed on or close to the inner wall surface. A reasonable amount of moisture-protected insulation should be used to minimize heat losses and installed so as to provide an air space within the wall. The air space offers little heat resistance but it may offer considerable insurance against harmful effects resulting from moisture deposition. Venting of the stud space to the low temperature side which is a region of low vapor pressure will also assist in preventing deposition of moisture. By observing such precautions, the difficulties encountered will be mitigated.

## Tons of Coal in Loose Piles

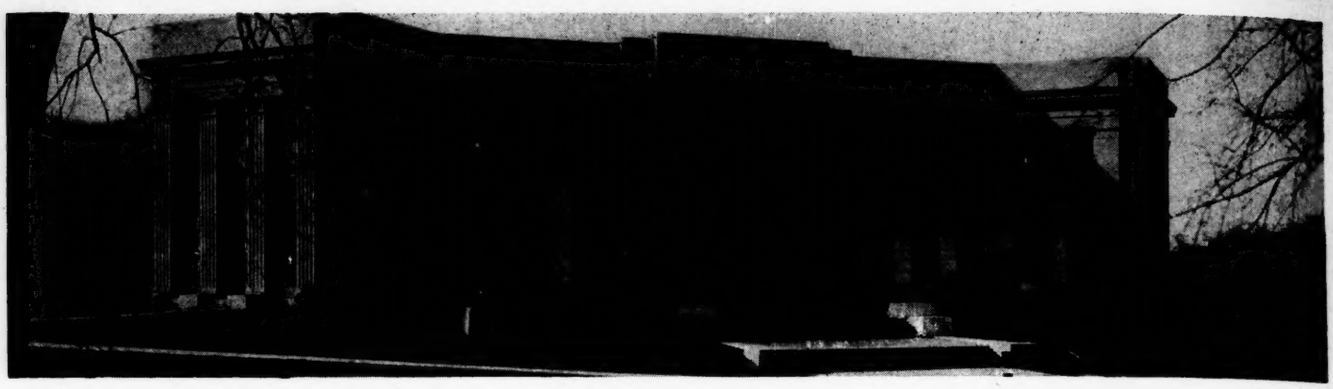
WHEN bituminous coal is piled in a conical pile the tonnage in the volume is easily estimated from the formula $T = 0.00248D^3$ where $T$ is the number of tons and $D$ is the diameter of the pile in feet at its base.

For anthracite coal the formula is $T = 0.00172D^3$.

But coal is often stored in elongated piles, in which case the tonnage of bituminous coal is computed from the formula $T = 0.00248D^3 + 0.00474AD^2$ where $A$ is the length of the "crest" on top of the pile in feet. $D$ is the width in feet of the base of the elongated pile.

Where the coal in the elongated pile is anthracite instead of bituminous the formula becomes: $T = 0.00172D^3 + 0.0033AD^2$.

$D$ and $A$ are both measured in feet in all of these formulas which are based on the fact that one ton of coal occupies approximately 38 cu. ft. on the average, whether anthracite or bituminous. The natural cone of anthracite is 0.25 as high as it is in diameter at its base. For bituminous coal the height factor is 0.36.



# 12-inch Roof Pool Insulates Building

A SPLIT duct system for the air conditioning and a roof pool for insulation are features of the new Alabama State Highway Building, erected at a cost of $450,000 in Montgomery.

The architectural design of the building itself is unusual. Concrete monolithic construction was chosen since it was a WPA project and it was desired to give as many man-hours of work as possible. Availability of sand, gravel and cinders was also a determining factor.

Main equipment for the 160-ton air conditioning installation is placed in the basement with heat being provided by two cross-connected steam boilers, each served by a coal stoker. Separate fan units serve each floor, the units for the basement and first floor being located in the basement and for the second and third floor in the attic.

By means of the split ducts, tempered air is passed through one half and either hot or cool air through the other half. By means of a hand-operated damper at the outlet grilles, room occupants may regulate the temperature to suit. The damper closes one side or the other of the duct at a time. In addition each take-off duct has an adjustable valve by which the predetermined temperature of any room may be regulated to suit the occupants.

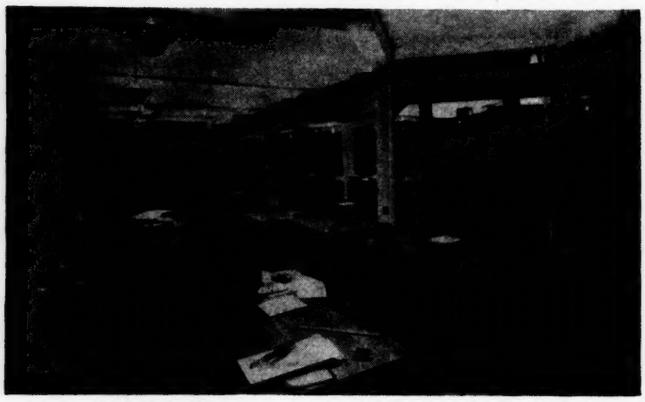
According to C. P. Lichty, of C. P. Lichty Engineering Co., Inc., who designed the air conditioning, the split ductwork will help solve the problem whereby some occupants of the building think the temperature too hot and others too cold. The flexibility of the system will enable it to serve the drafting room where the lighting intensity is high and more heat is generated, also the testing laboratory, auditorium where highway delegations are received, private offices and conference rooms.

Main ductwork is furred in against the hallway ceilings and insulated with one inch of cork. Cinder concrete blocks used for the building partitions provide a degree of insulation for the conditioned spaces as well as acoustical properties. Return air is handled through louvers in the hallway doors. Water from a well on the premises serves the air conditioning system.

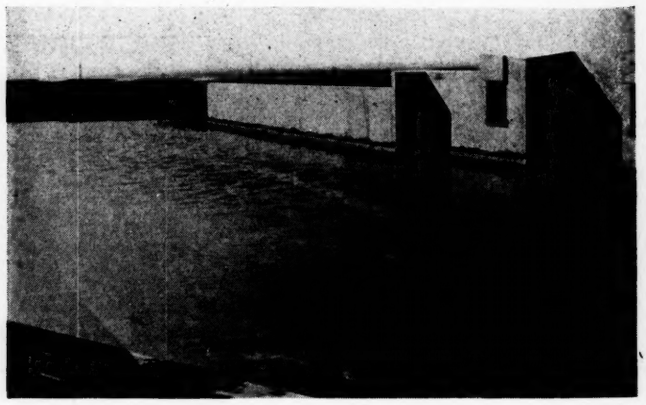
Vent shafts for the laboratory and toilet rooms in the building were provided in an unusual manner by placing steel or concrete pipes inside the concrete masonry of the columns and pilasters of the building itself.

The roof pool was the idea of H. H. Houk, chief highway engineer, who had charge of the design and construction of the building. The plan is to keep a 12-in. sheet of water on the roof winter and summer. It will also help to provide a more even temperature for the upper floors, particularly for the attic which is used for storage of records and documents and is not air conditioned.

This pool is serviced by 40 supply lines of 1½-in. copper placed in the retaining walls. Each line has a spray head. A number of roof floor drains are provided with 4-in valves accessible in the attic. — *George H. Watson.*



*Drafting department, showing ductwork.*



*Roof pool for insulation.*

# • EDITORIALS •

## Promising Data from Hospitals

Last month we published results obtained at Corey Hill Hospital in Boston where a most interesting air conditioning installation has been operating for nearly a year. Not only did the equipment perform suitably but the hospital management was able to collect data showing how the patients reacted and how the conditioning affected them. Field results from new types of installations are always of interest but we believe that they are especially so in the case of this hospital. Much is hoped for when air conditioning is generally applied to hospitals, for there is some reason to believe that it may exert more potent effects than are generally conceded to it now. There is little reason to expect that hospitals generally will install air conditioning until there is enough experience to prove both its benefits and its problems. While the results at Corey Hill are among the early ones they should be only the forerunners of many more to come for enough hospitals now have installations so that the experiences should be available before long.

## Change in Naming Gas Heaters

In a recent issue of AGA Monthly, publication of the American Gas Association, a writer calls attention to the desirability of substituting the words "direct heating" for "space heating" as being more acceptable to the people who apply gas-fired heating equipment. He also states that they "usually use the term 'space heating' to designate the heating of any given space, whether it is an office building or a modest dwelling." This is doubtless a fact, and the terms "space heating" and "space heaters" have not been particularly apt or widely accepted outside the gas industry itself. It appears that a change in phrasing is desirable.

Generally, however, the term "direct heating" has an already well defined and accepted meaning to those engaged in heating and ventilating applications. To them it means heating where the source of heat is located in the heated space. They make a well defined distinction between direct and indirect heating. It would seem desirable that in naming its equipment the gas group would do well to follow the lines already pretty well laid out by common usage.

## Sales Taxes for Us?

With the search for tax sources keen there are few possibilities which the tax gatherer overlooks. Repeatedly the eye has been turned toward anything which could be classed as a luxury. With this true it is not surprising that heating and cooling equipment above the bare minimum has been examined, as have also the fuels used for heating. So far these examinations have not resulted in the laying of taxes but there is no assurance that this situation will continue. A Federal sales tax on oil used for heating purposes is now receiving attention. Here and there suggestions have been made that air conditioning equipment might well bear a sales tax. Nothing is seriously expected to eventuate from these proposals but all such suggestions should be fought both on the grounds that sales taxes generally are not equitable and that the imposition of sales taxes would provide a burden which would seriously hamper developments in places where development is now going on steadily.

## New Housing Act

The newest housing act was passed and signed with scarcely more than a ripple of interest in the building industry. Likewise the other administration suggestions to boost building activity have been received with a show of indifference. Whether this indifference is due to a feeling that the act and the suggestions will produce but a few results is a point which cannot be determined. At any rate there was nothing resembling the enthusiasm shown when the original housing act for insuring loans on modernization work was enacted. The new act provides for a revival of the insurance feature and in effect puts FHA back into the picture. It also extends the limit of insured loans on new construction up to 90% instead of fixing the limit at 80% as formerly. This means decreasing the down payment presumably, although it remains to be seen whether lenders will extend loans up to the limit or not. Advance opinions are divided as to what the practical effects will be. To a considerable extent there is a tendency to adopt a wait-and-see attitude. In the meantime little is being heard about the proposal that labor accept an annual wage based on lowered hourly rates. It is probably correct to conclude that the building industry does not have any great hopes that it will receive much aid from the government program. Perhaps this is a good mental attitude, for if we expect but little we can be pleasantly surprised if the program does result in stimulating building.

## Clean Air for Comfort

At a recent meeting a man long connected with the development of air cleaning equipment remarked on how relatively little recognition is given to the problem of removing dirt and dust from air as compared to the problem of removing heat and moisture from it. To a thoughtful person this is truly rather puzzling. By and large there is probably more actual evidence in existence whereby it can be shown that air dust and similar impurities should be removed in preference to heat or humidity. Still, for the usual comfort application more money is spent on heat and moisture removal than on cleaning. The answer must lie to some extent in the greater ease with which heat and moisture removal can be dramatized and capitalized. Also heat and moisture removal affect the feeling of comfort quickly and demonstrably, while the effects of dust and dirt are slow-acting and not so evident over short periods. Even without the advantage of dramatic appeal the necessity of providing clean and wholesome air for comfort is being steadily recognized.

# NEWS OF THE MONTH

## Sherman, Marsh, and Riecks Address 22nd Fuel Conference Held in Atlanta by Appalachian Coals, Inc.

ATLANTA—The 22nd fuel engineering conference sponsored by Appalachian Coals, Inc., was held here February 8. Speaking to 243 persons, who registered for the sessions, well-known stoker, fuel, and power authorities discussed problems of interest to industrial executives, fuel engineers, retail coal merchants, and stoker dealers.

Thirteen states — Alabama, Georgia, Illinois, Kentucky, Massachusetts, Michigan, New Jersey, North Carolina, Ohio, Pennsylvania, South Carolina, Tennessee, and Virginia—and the District of Columbia were represented at the conference, with the largest groups in attendance being from the several southeastern states.

W. H. Riecks, Detroit, secretary-treasurer, Detroit Stoker Co., read a paper on The Spreader Stoker, prepared by M. H. Snodgrass, mechanical engineer, Detroit Stoker Co., who was unable to attend. This paper discussed some of the many applications of spreader stokers. Nearly 50 slides, giving installation views and drawings showing application to various types of boilers, were used to illustrate the paper.

The spreader stoker, the paper stated, offers a solution for many combustion problems. It is a system for burning fuel for steam boilers and has been thoroughly tested with various fuels. The spreader stoker successfully burns comparatively inexpensive sizes of coal, such as 1¼- or ¾-in. screenings, or straight slack, without preparation. The fuel is automatically fed from the hoppers to revolving rotors, which throw it over and into the furnace through an overfeed action. The spreader type stoker burns the fine particles of fuel in suspension and the coarse particles on grates of special design. It is extremely flexible and responsive to sudden changes in load. . . . The cleaning of fires is a comparatively simple operation with the spreader stoker.

T. A. Marsh, Chicago, central division engineer, The Iron Fireman Manufacturing Co., discussed Small Industrial and Residential Stokers. Among other things, Mr. Marsh pointed out that the stoker industry had grown rapidly, even though its products are relatively new; that sales of stokers had doubled each year for five years, including the depression period.

The speaker used a colored motion picture to describe the operation of a stoker designed for commercial installation. This picture gave the retailers and the stoker dealers a new conception of what automatic coal burning equipment will do.

Automatic Residential Heating with Bituminous Coal, by R. A. Sherman, supervisor, Battelle Memorial Institute, concluded the conference.

Mr. Sherman said that the demand of the people of this country for automatic heating results from their desire for convenience, comfort, health, and luxury. It conforms with their demand for the automobile, mechanical refrigeration, and the numerous other labor-saving devices for the home. That people do want and will buy automatic heating equipment has been proved by the sale of oil-fired and gas-fired heating equipment which preceded the coal stoker in development and in public acceptance.

The small underfeed stoker, he concluded, has proved itself to be an eminently satisfactory method of heating homes automatically. It possesses the essential merits of convenience, cleanliness, quietness, and dependability and, in most parts of the country, gives these advantages with the lowest cost source of heat, bituminous coal.

The coal industry, producers, wholesalers, and retailers, owe a debt to the manufacturers of these stokers who have given the coal industry this means of combating the encroachment of the competitive fuels. Not only are furnaces and boilers that have been hand fired with coal being converted to stokers and thus saved to coal, but oil burners and gas burners are being replaced by stokers. Even if the stokers were replacing only hand firing, there is much comfort in the knowledge that a potential user of gas and oil is saved for coal. To paraphrase Ben Franklin, "A coal user saved is a coal user gained."

The stoker manufacturers, Mr. Sherman said, have had a difficult problem to adapt their equipment to the many types of bituminous coals that are found. They have done a splendid job and there are few coals today that will not give satisfaction. Their efforts to make the stoker truly universal for all coals is still going on and we can shortly expect it to be solved. A solution that includes completely automatic operation with feeding of coal from a bin, which is now wholly satisfactory, and with automatic removal of the ash, will surely come in the near future.

## L. A. Brady to Assist Porter at OBI

NEW YORK—L. A. Brady, formerly executive secretary of the Fuel Oil Association, Inc., of New York, has joined the staff of Oil Burner Institute as assistant to G. Harvey Porter, managing director. Mr. Brady will have charge of certain regional activities.

## Industry Aids Air Hygiene Studies; Start Drive on Occupational Disease

PITTSBURGH—Fifteen representatives of industry, science, and government, were elected to the board of trustees of Air Hygiene Foundation at the annual business meeting held at Mellon Institute, February 8. H. B. Meller, managing director, announced in his annual report that 200 industrial concerns are now supporting the Foundation's research and studies for the prevention of occupational disease. Meller said this basic work, "of vital human and economic import," is progressing at several research centers, including The Saranac Laboratory, the Moore School X-Ray Laboratory at the University of Pennsylvania, and at Mellon Institute.

## N.Y. Contractors Hold Annual Dinner

NEW YORK—The Association of Plumbing and Heating Contractors of Greater New York, Inc., held the largest social affair in its history February 21 when members and guests attended the annual dinner of the Association at the International Casino. The attendance was above all expectations.

The Association last year changed its name to include plumbing and has moved its headquarters to 304 East 62nd St. The annual election was held during January and the entire slate was reelected as follows: Philip Wolf, president; J. Kessler, vice-president; B. Spiegel, treasurer; J. C. Weare, secretary; J. Miller, chairman of the Board of Directors, and Prof. F. W. Hanburger, consultant.

Heating and plumbing contractors operating in Greater New York are eligible for membership. The dues are $10 per year with no assessments.

## Adlam Describes Radiant Heating

CHICAGO—The fourth meeting of the ASHVE Illinois chapter's season was held January 10 with 96 present at dinner and 135 attending the meeting.

Panel Heating was the subject of the evening and it proved of absorbing interest to the members. T. Napier Adlam, chief engineer, Sarco Co., Inc., was the speaker and also on the program was Irving Brooke, consulting engineer, who had with him Denison B. Hull, architect and owner of the only home in the Chicago district with panel heating, a job designed by Mr. Brooke.

Mr. Adlam discussed the theory of radiant or panel heating, and showed a number of slides which pictured the equipment used, methods of installation, calculation of surface requirements, and examples of panel heating in a variety of buildings.

## Electric Air Cleaner in New 5 & 10

NEW YORK—At a luncheon in the Engineers' Club, representatives of the press saw a demonstration of the electrical air cleaner developed by Westinghouse Electric & Manufacturing Company as applied to the F. W. Woolworth & Company's new modern store at 39th Street and Fifth Avenue. The store was formally opened February 23.

With a graphic chart depicting air impurities magnified 5000 times, Carl P. Bernhardt, Westinghouse engineer, showed how dust particles three times as large as ragweed pollen pass easily through a 200-mesh screen as tightly woven as the finest dress fabric.

Then J. H. Squires illustrated with a model of the Woolworth machine how the electric air cleaner not only removes ragweed pollen, but also tobacco smoke, particles of which could pass through a 200-mesh screen 576 abreast.

He noted that in the area surrounding the new Woolworth store, dirt is deposited out of the atmosphere at the rate of 1000 tons per square mile annually. Nevertheless, he said, inside the store the store the cleaner would constantly remove 99%, by weight, of all particles in the atmosphere, even those as small as four millionths of an inch in diameter.

## Dust and Fume Codes Under Way

BOSTON—To assist the Massachusetts Department of Labor and Industries to draft dust and fume codes, Commissioner James T. Moriarty has appointed the following committee: Dr. Joseph Aub, Huntington Memorial Hospitals; John Daniels, legislative agent, Associated Industries of Massachusetts; William M. Rand, vice-president, Monsanto Chemical Co. and president of A.I.M.; Prof. Philip Drinker of Harvard; E. H. Ballard, General Electric Co.; Laurence Foley, Granite Cutters' International Association; Dr. Francis T. Hunter, Massachusetts General Hospital; Robert J. Watt, Massachusetts Federation of Labor; and Stephen E. Whiting, Liberty Mutual Insurance Company.

## Mountain to Have New Observatory

MT. WASHINGTON, N. H.—The famous weather observatory on the summit of Mount Washington has just moved into a sturdy building of its own, built to withstand a 200-m.p.h. wind. Such a wind occurred in 1934, and caused the former building to bulge in and out like a bellows because of pressure variations of as much as two-tenths of an inch by the barometer.

Heat is supplied from a warm air furnace and hot water from a 2500-gal. tank inside the building, filled with water from the base station by a pump.

## Contractors Organize in LosAngeles

LOS ANGELES—Formed in southern California last September, the Heating, Piping, and Air Conditioning Contractors Association has been making rapid progress.

It is the purpose of this group to sponsor Certified Heating and Ventilating and committees have been formed to call on engineers and architects and mechanical engineering departments of the city and county to discuss ways and means to carry out this program. Plans are also being formulated to form a bid depository. This form of bidding is in effect and force in practically all the major cities in California and is operating successfully.

The officers of the Association are: J. M. Moriarity, president; C. F. Ballway, vice-president; Frank J. White, secretary; and B. K. Stoneman, treasurer. The board of directors consists of Andrew Felthouse, J. Herman, Lloyd Johnson, James Yale, B. K. Stoneman, C. F. Ballway and F. J. White.

All heating and ventilating contractors of Southern California are invited to join this Association, the offices of which are located in the Beaux Arts Building here.

## Student-Built A.C.Units to be Shown

SAN LUIS OBISPO, CALIF.—Students of California State Polytechnic have completed their plans for the exhibition of their project work for the 6th Annual Poly Royal Show which is to be held April 1-2 on the State Tech campus.

One of the highlights of the show is to be the air conditioning unit which is being completed by the students of the air conditioning department. Several unit heaters built by the students will also be on display.

The California State Polytechnic is the only state school in California that has a two-year vocational course in air conditioning, ventilating, and refrigeration.

## Gurney Began Heating Career at 14

TORONTO, ONT.—E. Holt Gurney, president of the ASHVE, started his business career at 14 when he sold stoves in a retail store. Mr. Gurney's grandfather founded the Gurney Foundry Co., Ltd., in 1843, and at his death the administration of the business passed to Mr. Gurney's father. E. H. Gurney is therefore the third generation in charge of the business which now has plants in Toronto and other Canadian cities. Air conditioning equipment is one of the firm's featured products.

Mr. Gurney, who is a graduate of the University of Toronto, was a championship runner in his early youth. Today he is a splendid tennis player and golfer.

## New Yorkers Discuss Steam Jet A.C.

NEW YORK—Steam jet refrigeration as applied to air conditioning was the subject of the regular monthly meeting of the New York chapter of the ASHVE February 21. The speakers of the evening were A. R. Mumford and A. A. Maxon, research engineer and assistant research engineer, respectively, New York Steam Corp.

The speakers showed that if steam jet refrigeration for air conditioning were applied to 10% of the space now served by district steam in New York, the New York utility's load factor, or ratio of the average month to the peak month, would increase from 47.35% to 51.6%, while if 20% of the space were conditioned the annual load factor would be 56.0%. The purpose of the speakers' talk was to demonstrate and discuss air conditioning applications which fall into the economical range of water vapor refrigeration; to show that steam jet refrigeration should be treated as a direct process in order to obtain real operating advantages in economies; to familiarize engineers with the load characteristics of steam jet apparatus, and to discuss condenser water from a constructive engineering viewpoint.

They analyzed steam jet refrigeration and explored its limitations theoretically and then applied some of their findings to steam jet installations operating here. They pointed out that where central station steam is available in Manhattan the operating costs of steam jet are such that it can not be disregarded in any economic setup.

Following their talk Charles Hoffman, chief engineer, First National Bank Building, spoke regarding some of his operating experiences where steam jet equipment is being used. O. L. Maddux, a visitor from Hamilton, Ontario, also discussed his findings, pointing out that the observed results agreed with the work done by Director F. C. Houghten at the ASHVE research laboratory.

## Drinker Addresses Toronto Group

TORONTO, ONT.—The speaker at the February meeting of the Ontario chapter, ASHVE, held in the Royal York Hotel, was Prof. Philip Drinker, associate professor of industrial hygiene at Harvard. Professor Drinker spoke on "Air Pollution and Dust Control," a subject on which he is regarded as an authority.

Vice-president H. B. Jenney acted as chairman of the meeting in the absence of President G. Playfair.

T. MacDonald presented Harry R. Roth, with a travelling bag as a gift from the chapter, in appreciation of the work he has done as secretary of the Ontario chapter for the past few years.

Fifty-three attended the meeting.

## Experts Talk on Fire Hazards of A.C. at Chicago Meeting

CHICAGO—The February meeting of the Illinois chapter, ASHVE, was held at the Brevoort Hotel February 14, with 82 members and guests present.

President Rottmayer introduced J. T. W. Babcock, engineer, Western Factory Insurance Association, who spoke on fire hazards in connection with heating, ventilating, and air conditioning. He pointed out that it is a matter of history that almost every engineering development is accompanied with its era of fires and that air conditioning has presented new problems which must be met. He stated that safeguards begin on the drawing board of the designer and urged the cooperation of the engineers in eliminating the hazards which improperly designed systems might set up.

Mr. Babcock was followed by R. E. Maginnis, special agent, American District Telegraph Co., who described methods of protection against fires.

It was apparent from the two talks that insurance companies are giving careful study to air conditioning and that the engineers should know what regulations are being prepared and should cooperate to the best interests of all concerned.

## Armour Tech Sponsors Conference

CHICAGO—Armour Institute of Technology announces that it will sponsor the Midwest Power Conference to be held here April 13-15. Six midwestern state universities will cooperate in operating the conference which will replace the original one organized under private and commercial sponsorship.

The program will involve some thirty papers by authors drawn from educational and industrial fields and divided among steam, diesel, electric and automatic power.

Power requirements for air conditioning will be discussed on Friday morning, April 15.

Registration and headquarters will be at the La Salle Hotel. Further information can be obtained from L. E. Grinter, Dean of the Graduate Division, Armour Institute of Technology, Chicago.

## St. Louis Contractors Elect Smith

ST. LOUIS—At the regular meeting of the Heating, Piping and Air Conditioning Contractors St. Louis Association, held at its building, 3626 West Pine Boulevard, February 2, W. M. Smith was elected president.

Other officers elected for the coming year were: vice-president, John Meara; secretary, James Fagan; treasurer, Jules E. Strong; sergeant-at-arms, Theodore Karr; trustees, Edwin P. Ambler, Bernard Braun, and Charles K. Ramsey.

## Resist Discriminatory Tax on Oil, Dewey Urges Burner Men

DETROIT—In an interview February 9, Frank H. Dewey, new president of OBI, said that the most important needs of the oil burner industry today are: (1) An advertising campaign by the oil companies to match the national gas and coal drives to dominate the automatic heating market, to the disadvantage of oil burning equipment; (2) organized cooperation in resisting discriminatory taxation on fuel oil, promoting uniform ordinances and expediting proper action on unfavorable legislation, and (3) establishing of standardized practices with relation to installation, service, parts and financing, and the improving of relations among manufacturers, dealers and oil marketers.

President Dewey expects to concentrate his personal attention on the first of these needs—a national oil heat advertising campaign by the oil companies—and to the corralling of all responsible burner manufacturers in the United States into the membership of OBI.

In a letter recently sent to a number of manufacturers not now members of the Institute he said, that "... the unsettled conditions existing today in the oil heating and air conditioning industry are a source of concern to all of us who manufacture equipment. These conditions will be improved when all responsible companies join in a concerted movement for good of the industry. No one of us can do it alone."

President Dewey urged early and concerted action on each of the three industry needs outlined and pledged the effort of the Institute and its staff toward these objectives.

## Monroe Addresses Boston Bureau

BOSTON—Members of the Air Conditioning Bureau and invited guests met at the Edison Auditorium February 16 for the monthly meeting. L. E. Monroe, secretary of the National Association of Fan Manufacturers, was the guest speaker, taking for his subject "Fan Standards and Applications as Developed and Approved by the Industry."

## Charles Allensby

LONDON—Charles Allensby, president of the British Institute of Heating and Ventilating Engineers, died recently from injuries received in a railroad wreck near Bedford late in January.

Mr. Allensby was travelling to the annual dinner of the East Midlands branch of the Institute when the accident occurred. The members, on learning of the crash, abandoned the dinner.

Mr. Allensby, who was 56, was a director of Benham and Sons, London kitchen and heating engineers. He leaves a wife and three daughters.

## 8th Warm Air Course Scheduled at Michigan State College

EAST LANSING — The eighth annual forced warm air short course will be held at Michigan State College here March 28-31.

The program is arranged in three parts: (1) For advanced heating students, under the direction of Fred Bishop and Ed Root with Gilbert Denges and Frank Meyer; (2) for beginners under the direction of Bruce McLouth, and (3) for students interested in calculating a cooling job, under the direction of W. G. Schlichting.

Among the papers to be presented at the general session are Seasonal Heat Loss, by C. Alder; Soil Temperatures and Heat Loss Below Grade, by C. H. Pesterfield; Recen' Developments in Air Cleaning, by H. C. Murphy; The Humidity-Insulation Problem, by L. G. Miller; Automatic Controls for the Residence, by C. W. Nessell.

Among other subjects treated for advanced heating students will be Special Problems in Figuring Heat Loss, by Fred Bishop, and Duct Layout and Design, by Ed Root.

Mr. McLouth will direct lectures for beginners on Heat Losses, C.f.m. Requirements, The Layout, etc.

Mr. Schlichting will lecture for students calculating a cooling job on Heat Gains, Heat Gains Converted to Cooling Load, Air for Cooling, Cooling Problems, and Selection of Cooling Equipment. For these students S. H. Downs of Clarage Fan Company will discuss methods of Fan Selection.

## J. P. Eddie

LANSING, MICH.—J. P. Eddie, sales manager of Dail Steel Products Co., died early in February following an illness of nearly a year. Associated with Dail for 17 years, Mr. Eddie was formerly with the New-Way Motor Co. He was born in Canada 50 years ago. Shortly after coming to Lansing he was connected for several years with the Pere Marquette Railway. He also served eight years on the Electric Light and Water Commission of Lansing.

## Horace E. Walters

BIRMINGHAM, ALA.—Horace E. Walters, 49 years old, district manager of Crane Company's branch in this city, died suddenly February 23 in Chicago. He was attending a meeting of district managers in that city.

Joining Crane's Atlanta, Ga., branch in 1911, Mr. Walters was later transferred to Savannah and then to New Orleans. In June, 1937 he was appointed district manager in charge of the company's southeastern branches.

Mr. Walters is survived by his widow and two sons.

## TABLE 1—HEAT EMISSIONS OF BARE, UNENCLOSED, DIRECT CAST IRON RADIATORS
(70F Air and 215F Surface Temperature)

### Column Radiators[1]

| Height, Floor to Top, Inches | Number of Columns | | | | | | |
|---|---|---|---|---|---|---|---|
| | 1 | 2 | 3 | 4 | 5 | 6 | Solid Type |
| | Heat Emission (H) per Section, E.D.R. | | | | | | |
| 12 | — | — | — | — | — | 3.00 | — |
| 13 | — | — | — | — | 3.00 | — | — |
| 14 | — | — | — | — | 3.00 | 3.25 | — |
| 15 | — | 1.50 | — | — | — | — | — |
| 16 | — | — | — | — | 3.50 | 3.75 | — |
| 17 | — | — | — | — | — | 4.00 | — |
| 18 | — | — | 2.25 | 3.00 | 4.00 | 4.25 | — |
| 20 | 1.50 | 2.00 | — | — | 5.00 | 5.25 | 3.25 |
| 22 | — | — | 3.00 | 4.00 | 6.00 | — | — |
| 23 | 1.67 | 2.33 | — | 4.67 | — | — | 4.00 |
| 26 | 2.00 | 2.67 | 3.75 | 5.00 | 7.00 | — | 4.50 |
| 28 | — | — | — | — | — | — | 5.00 |
| 32 | 2.50 | 3.33 | 4.50 | 6.50 | 8.50 | — | 5.75 |
| 38 | 3.00 | 4.00 | 5.00 | 8.00 | 10.00 | — | 7.00 |
| 45 | — | 5.00 | 6.00 | 10.00 | — | — | — |

[1]Of the type manufactured prior to 1926. These figures are for ordinary radiators in the field, and not for design purposes. For design work, see manufacturers' ratings.

### Tubular Radiators[2]

| Height, Floor to Top, Inches | Number of Tubes | | | | |
|---|---|---|---|---|---|
| | 3 | 4 | 5 | 6 | 7 |
| | Heat Emission (H) per Section, E.D.R. | | | | |
| 13 | — | — | — | — | 1.50 |
| 14 | — | — | — | — | 2.00 |
| 16 | — | — | — | — | 2.50 |
| 17 | — | — | 2.25 | — | 2.67 |
| 18 | — | — | — | — | 2.83 |
| 20 | 1.75 | 2.25 | 2.67 | 3.00 | 3.17 |
| 23 | 2.00 | — | 3.00 | — | 3.50 |
| 25 | 2.00 | 2.50 | 3.00 | 3.50 | 4.00 |
| 26 | 2.33 | 2.75 | 3.50 | 4.00 | 4.50 |
| 29 | 2.50 | 3.25 | 4.00 | 4.50 | 5.00 |
| 31 | 2.50 | — | 4.25 | — | — |
| 32 | 2.50 | 3.50 | 4.25 | 5.00 | 5.75 |
| 36 | 2.50 | 3.75 | 5.00 | 6.00 | 6.50 |
| 37 | — | 4.00 | 5.00 | 6.00 | 7.00 |
| 38 | 3.00 | 4.25 | 5.00 | 6.00 | 7.00 |

[2]Of the type manufactured since 1926. For ordinary radiators in the field, and not for design purposes. For design work, see manufacturers' ratings.

### Ultra-Slender Tubular Radiators

| Height, Floor to Top, Inches | Number of Tubes | | |
|---|---|---|---|
| | 3 | 4 | 5 |
| | Heat Emission (H) per Section, E.D.R. | | |
| 19 | 1.1 | 1.4 | 1.8 |
| 22 | 1.3 | 1.6 | 2.1 |
| 25 | 1.5 | 1.8 | 2.4 |

### Bathroom Radiators

| Short Side, Inches | Long Side, Inches | |
|---|---|---|
| | 12 | 20½ |
| | Heat Emission (H), E.D.R. | |
| 8 | — | 3.00 |
| 9 | 3.00 | — |
| 12 | 4.25 | — |

### Flat Front Wall Radiators[3]

| Number of Tubes | Height, Inches | | |
|---|---|---|---|
| | 17 | 20 | 25 |
| | Heat Emission (H) per Section, E.D.R. | | |
| 1 | 1.75 | 2.00 | 2.25 |
| 2 | 3.33 | — | 4.25 |
| 3 | 3.00 | 3.33 | 3.67 |
| 4 | 3.67 | 4.00 | 5.33 |
| 5 | 5.25 | — | 6.00 |

[3]Of the type known as Paxton, Pangi, etc.

### Wall Radiators

| Dimension, One Side, Inches[4] | Distance, Other Side, Inches[4] | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | 16½ | 18½ | 21 | 22 | 25 | 26 | 28 | 29 | 30 | 32 |
| | Heat Emission (H) per Section, E.D.R. | | | | | | | | | |
| 3½ | — | — | 1.5 | — | — | — | 1.8 | — | — | — |
| 12 | — | — | — | 5.0 | — | — | — | 7.0 | 8.0 | — |
| 12½ | 5.0 | — | 6.0 | 7.0 | 7.5 | — | — | 8.0 | — | — |
| 13 5/16 | 5.0 | 7.0 | — | 7.0 | — | 8.0 | — | 8.0 | — | — |
| 14½ | 5.0 | — | — | 7.0 | — | — | — | 8.0 | — | — |

[4]Dimensions and emissions approximate only.

## TABLE 2—RELATIVE HEAT EMISSIONS OF BARE CAST IRON RADIATORS FOR VARIOUS RADIATOR AND ROOM TEMPERATURES

| Room Temperature, F | Surface Temperature, F | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | 150 | 160 | 170 | 180 | 190 | 200 | 210 | 215 | 250 | 275 | 300 |
| | Multipliers (F) For Bare Cast Iron Radiators | | | | | | | | | | |
| 40 | 0.70 | 0.78 | 0.87 | 0.96 | 1.04 | 1.13 | 1.22 | 1.30 | 1.64 | 1.83 | 2.01 |
| 50 | 0.62 | 0.70 | 0.78 | 0.87 | 0.96 | 1.04 | 1.13 | 1.30 | 1.64 | 1.83 | 2.01 |
| 60 | 0.54 | 0.62 | 0.70 | 0.78 | 0.87 | 0.96 | 1.04 | 1.13 | 1.48 | 1.66 | 1.83 |
| 65 | 0.50 | 0.57 | 0.65 | 0.74 | 0.83 | 0.92 | 1.00 | 1.09 | 1.43 | 1.60 | 1.78 |
| 70 | 0.45 | 0.54 | 0.62 | 0.70 | 0.78 | 0.87 | 0.96 | 1.00 | 1.35 | 1.57 | 1.72 |
| 75 | 0.41 | 0.50 | 0.57 | 0.65 | 0.74 | 0.83 | 0.92 | 0.96 | 1.30 | 1.54 | 1.70 |
| 80 | 0.39 | 0.46 | 0.54 | 0.62 | 0.70 | 0.78 | 0.87 | 0.92 | 1.26 | 1.50 | 1.65 |
| 90 | 0.32 | 0.39 | 0.46 | 0.54 | 0.62 | 0.70 | 0.78 | 0.83 | 1.17 | 1.39 | 1.54 |
| | 22.4 | 20.3 | 17.7 | 14.6 | 10.9 | 6.4 | 1.8 | 1 | 15 | 30 | 45 |
| | Steam Gage, Vacuum, In. of Mercury | | | | | | | Steam Gage, Pressure, Lb. | | | |

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# ESTIMATING THE HEAT GIVEN OFF BY DIRECT C. I. RADIATORS

**TABLE 3—RELATIVE EFFECT OF RADIATOR FINISHES ON HEAT EMISSION**

| Finish | Multiplier (P) |
|---|---|
| Bare cast iron, foundry finish | 1.00 |
| Gold bronze | 0.90 |
| Aluminum bronze | 0.91 |
| White paint | 1.02 |
| Cream paint | 1.04 |
| Red lacquer | 1.00 |
| Dull green paint | .96 |
| Brown paint | 1.05 |

**TABLE 4—RELATIVE EFFECT OF ENCLOSURES ON RADIATOR HEAT EMISSION**

| Description of Enclosure | Multiplier (E) |
|---|---|
| None, bare cast iron radiator | 1.00 |
| None, shield above radiator | 0.90 |
| Open front and ends at bottom. Large free area in outlet, front only, upper half | 0.90 |
| Solid top, grilled front and ends, with large free area | 0.86 |
| Solid top, grilled front and ends, small free area | 0.85 |
| Solid top and ends, full grilled front. Factor depends on free area of grille; drops off rapidly as free area decreases | 0.70-0.87 |
| Solid top and ends, upper half of front grilled | 0.83 |

This sheet makes readily available the information necessary in the field to determine the net heat given off by direct cast iron radiators. The data given are not intended to be used in design but are for surveys of existing plants.

Each of the tables can be used alone if desired. To determine the combined effect of the different variables affecting the heat given off by radiators, refer to the formula given later.

*Table 1*—Heat emissions in equivalent square feet (E.D.R.)† of bare, unenclosed, direct, cast iron radiators standing in still air at 70F and with 215F heating medium temperature. This table applies to column, tubular, ultra-slender, bathroom, flat front, and wall radiators.

The figures given in this table cover the ratings per section, except bathroom radiators, and imply steam condensing ability of 240 B.t.u. per hr. when having a 215F heating medium temperature in 70F air.

*Table 2*—Gives the relative heat emissions of radiators under varying radiator and room temperatures. In general it may be assumed that the radiator surface temperature will be the same as the temperature of the heating medium. The table shows that, for example, a radiator with 210F steam in an 80F room will use 0.87 as much steam as one standing in 70F air using 215F steam. Similarly, a radiator supplied with hot water at 170F in 70F air has a heat output 0.62 as much as a steam radiator in still air at 70 with a 215F surface. The factors given in this table when multiplied by the ratings in Table 1 give the equivalent square feet (E.D.R.) for the room and heating medium conditions given.

*Table 3*—Painting of a radiator affects its heat output. This table gives the relative effect of commonly used finishes as compared with a bare cast iron radiator.

*Table 4*—Gives the relative effect of some common enclosures on radiators as compared with bare, unenclosed, cast-iron direct radiators.

The combined effect of all of these variables can be expressed as

$R = H \times N \times F \times P \times E$, where

$R$ = Net Heat given off in E.D.R. (240 B.t.u. per hr.).

$H$ = Rated E.D.R. as given in Table 1.

$N$ = Number of sections of the given radiator.

$F$ = Factor from Table 2 depending on the room and heating medium temperature.

$P$ = Multiplier from Table 3, for finish.

$E$ = Multiplier from Table 4 depending on the type of enclosure used.

*Example*—What is the net heat given off by a 12-section tubular radiator with 5 tubes, 22 in. high, under 1 lb. per sq. in. steam pressure? The radiator is painted with aluminum bronze, has a shield above it, and is in a room maintained at 80F.

*Solution*—We can find the net heat given off by substituting in the formula

$$R = H \times N \times F \times P \times E$$

The section on tubular radiators of Table 1 shows that $H$ (the rated E.D.R. per section) for a 5-tube, 22-in., bare cast-iron radiator is 3.00.

$N$ (number of sections) is given as 12.

$F$ (correction factor for room and surface temperature) for an 80F room temperature and 1 lb. per sq. in. steam pressure is 0.92 in Table 2.

$P$ (multiplier depending on the radiator finish) is 0.91 for an aluminum finish from Table 3.

$E$ (multiplier for the enclosure) is found in Table 4 to be 0.90 for a shield above a radiator.

Substituting these figures in the formula

$R = 3.00 \times 12 \times 0.92 \times 0.91 \times 0.90 = 27.1$ E.D.R.

Radiators which use hot water for a heating medium frequently have a lower E.D.R. per section because of the lower surface temperatures maintained. Hot water temperatures will range anywhere from 170F for a gravity installation to 220F or higher for a forced pressure system. Of course, when surface temperatures of 215F or over are encountered, the E.D.R. will be equal to or higher than the rating under standard conditions.

In estimating the net heat given off by a hot water radiator the formula given can be used in the same manner as in the previous example. The rating will then be obtained in E.D.R. of 240 B.t.u. per hr. However, hot water equipment is rated in hot water E.D.R. It is common practice to call the 240 B.t.u. E.D.R. a steam E.D.R. and to use 150 B.t.u. per hr. as the hot water E.D.R. Therefore, to convert the steam E.D.R. to the hot water E.D.R. it is necessary to multiply by 240/150 or 1.6.

*Example*—What is the net heat given off by a 10-section, 38-in., 3-column radiator operating with 180F water in 75F air, finished with cream paint and having an enclosure with a solid top and grilled front and ends with a large free area?

*Solution*—$R = H \times N \times F \times P \times E$

$= 5.00 \times 10 \times 0.66 \times 1.04 \times 0.86$

$R = 19.5$ E.D.R. (Steam)

$19.5 \times 1.6 = 47.2$ E.D.R. (Water)

†One E.D.R. = 240 B.t.u. per hr.

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# WHAT READERS SAY • • •

We call attention of readers to the necessity for signing letters sent for possible inclusion in this department. This also applies to letters suggesting that some question be given attention in these columns. While letters as published do not necessarily reveal the identity of the writer, letters are not published or requests given attention unless they bear the signature of the writer—*Editor.*

●

## Rain and Air

In your issue for November, 1937, on page 56, you state, in reply to a reader's question, that "rain dries the air." In my opinion, your answer applies to a special condition not contemplated by the questioner, and for him inaccurate and misleading.

It is incontestably true that the formation of rain or snow reduces the absolute humidity at the place where condensation occurs, and therefore "dries the air." But who except a physicist cares about the conditions at the remote and inaccessible altitudes where rain forms? It is entirely reasonable to assume that the inquirer is interested in conditions near the ground level where human comfort is affected. For such conditions, rain is comparable to the spray in an air washer, and its action depends upon 1—temperature of rain; 2—dewpoint of atmosphere. A third and complicating factor is the ground temperature, which may for a time vaporize the rain with sufficient rapidity to create a blanket of air with a dewpoint high enough to enable dehumidification by the rain. Under such circumstances there would occur a combination of humidifying, dehumidifying, and evaporative cooling processes that would be difficult to analyze. It would seem that for the conditions probably contemplated by the questioner, there can be no answer without a set of specific conditions.—*R. B., Boston, Mass.*

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## Electric Heat

In the October, 1937, issue of HEATING & VENTILATING, page 49, the article on "Electric Heating of Residences" makes some statements that call for comments.

The design data (b) assumes the house heating period from 6 a.m. to 10 p.m., or 16 hr. per day. In Bulletin No. 93, referred to, the same assumption is used as a basis for calculations and curves as to heating and storage requirements. While the supply of heat to a residence may be halted 8 hr., we in Seattle find as yet no way to prevent the house itself from losing heat during that time, and nowhere in the bulletin is any allowance made for bringing the house back to normal temperature after being allowed to cool for a third of a day. If the temperature is not allowed to drop from 10 p.m. till 6 a.m., 50% more heat will have to be supplied than the author allows. If the house is allowed to cool, the amount of heat necessary to bring the house back to temperature in the morning will have to be added. This heat for morning pickup will vary from 30% to 90% of the amount of heat required for continuous operation during the 8 hr., or will mean from 10% to 30% more heat required during the 24 hr. The figure 10% would refer to a very flimsy house with only a small amount of capacity for storing heat in the building material itself, and the figure 30% would refer to a house with a large amount of heat stored in the building material itself.

The design data (c) assumes the heat storage temperature range from 100F to 250F. With a room temperature at 70F, radiator capacity with 100, 120, 140, and 160F water is only 13, 25, 39, and 54% of capacity with water at 215F. I doubt that water below 140F will be of much value in producing comfort conditions in residences even in Seattle.

By making allowances for the two conditions mentioned above, the quantity of water needed to store the required amount of heat must be increased a considerable amount. Furthermore, all the tanks had best be insulated, a factor the author seems to have overlooked entirely. Taking all these items into consideration, $684 will probably not provide a very adequate heating plant for a Class A residence.

Professor Magnussen is to be thanked, however, for his presentation of the problem and for information about the duration of near-zero weather in Seattle.—*C. H. James, Engineer, Seattle Branch, Montag.*

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## Furnace Men Protest

The writer read with interest the article by W. T. Schepeler, Engineer of Construction and Maintenance, Board of Education, Niagara Falls, N. Y., in your February issue. We presume that his employers are paying him for writing articles of this kind. Naturally, they will want all of the other Boards of Education to know that heating costs in a school can be cut 76% by tearing out furnace plants and installing split steam systems.

At any rate it is a very clever piece of advertising for the steam interests and the beauty of it is that it costs them nothing. I never was in Niagara Falls and know nothing about this particular school. I would like to read the furnace-man's side of that story.

It might interest you to know that over here last year we had a similar experience in a building of approximately the same dimensions and same description. We can show just as good a fuel saving and, I will wager, a considerable saving in installation cost. In fact, just as satisfactory a job all around. All that Mr. Schepeler would have had to do was to repair one old furnace and recase it, using it to temper the air for ventilation. A smaller boiler and direct radiation would have done the rest.

I would not be surprised to learn that the old furnace system in that building could have been redesigned into a modern air conditioning system and have given very satisfactory results. Chances are that the Board of Education of Niagara Falls were sold with just such

*(Concluded on page 84)*

# ABSTRACTS of Current Papers, Books and Pamphlets

## Copper Tube Handbook

Copper tube has found wide use in the heating and plumbing industry in the past few years. In many installations its properties make it an ideal material for easy installation and long life. This brief handbook was prepared to acquaint the engineer, the contractor, and the architect with the properties of copper tube and to show them how to design and install systems using it.

The book is divided into three main sections, the first dealing with general information, the second with plumbing, and the third with heating. Included in the section on general information is a discussion of the physical properties of copper tube such as resistance to crushing, freezing, expansion and contraction, and water hammer. This section also includes information on the types and properties of copper tubes, the fittings available, and the art of making soldered joints.

The section on plumbing gives design and installation data for copper tubes, the layout of a plumbing installation and plumbing layout details. Design information on steam and one- and two-pipe hot water systems occupies the third section. The latter part of this section has four pages of illustrations of heating details.

[*"Copper Tube Handbook," by T. M. Thompson. Issued by Copper and Brass Research Association, 420 Lexington Ave., New York. Flexible cover; 8½ x 11 in.; 44 pages.*]

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## Introduction of New Industrial Products

To assist the efficient introduction of new industrial products and to reduce to a minimum avoidable economic waste in marketing, the Marketing Research Division of the Bureau of Foreign and Domestic Commerce has revised and made available a new edition of a check list which enumerates procedures to be employed when introducing a new product to the market.

Included is a check sheet devoted to direct marketing problems and one devoted to problems concerning the origination and production of a new industrial product. Each of these check sheets is so constructed that the user has a wide list of problems which must be solved for the successful marketing of new industrial merchandise. These problems which show in the stubs are checked against the various angles in which they will be considered in order to insure that no important question is overlooked. There is a total of 844 problems listed.

The check sheet on marketing is divided into nine sections including the marketing, buying habits of the market, firm relation to the market, competition, seller's price policy, channels of distribution, sales promotion, management of sales forces, and others which include a number of questions which do not logically fall under any of the heads named. There are 560 problems indicated on the chart.

In the text of the book a section is devoted to a detailed explanation with examples of use for each section shown on the check sheet.

It includes a seven-page bibliography of government sources of industrial market research material of basic value to industrial marketers.

[*"Check Sheet Introduction of New Industrial Products," Market Research Series No. 6. Published by the Bureau of Foreign & Domestic Commerce, Washington, D. C. Copies available from the above bureau at 10c each.*]

●

## Arc Welding Handbook

In order to keep apace with the rapid developments in the field of arc welding the "Procedure Handbook of Arc Welding Design and Practice" has been brought out in a new edition. The fifth edition is encyclopedic in scope and might be said to be the complete arc welding reference guide for designers, engineers, architects, and welding operators.

The new material in this edition includes information on the characteristics of the welding generator, weld symbols, welding construction, high-speed automatic welding, high temperature stresses, machine design, structural design, and new welding applications in the industrial field.

The eight sections of the book cover the following subjects: welding methods and equipment; technique of welding; speeds and costs for welding mild steel; structure and properties of weld metal; weldability of metals; designing for arc welded steel construction of machinery; designing for arc welded structures; and typical applications of arc welding in manufacturing, construction, and maintenance.

[*"Procedure Handbook of Arc Welding Design and Practice," published by The Lincoln Electric Co., Cleveland, Ohio. Flexible binding; 6 x 9 in.; 1012 pages; price, $1.50 in the United States, $2 elsewhere; postage prepaid.*]

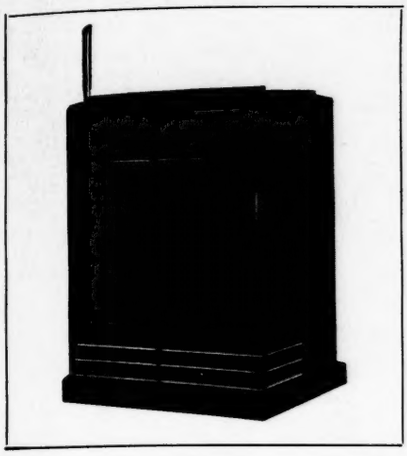
●

What Air Conditioning Means to the Water Utility. Presents information on the water requirements of air conditioning and gives data on the percentage of population for which air conditioning has been installed in 18 large cities. Also water consumption in these 18 cities based on a 60% application of conserving equipment. [*"What Air Conditioning Means to the Water Utility," by L. L. Lewis; Journal of the American Water Works Association, 29 W. 39th St., New York; June, 1937; pages 826 to 831.*]

Solar Energy for Heating Water. Presents an analysis of the use of solar energy for heating water in California and reports the results of investigations on water temperatures and the rate of heating water in different solar heating systems. [*"Solar Energy and Its Use for Heating Water in California," by F. A. Brooks. California State Bulletin 602 (1936), published by University of California, Berkeley Calif.*]

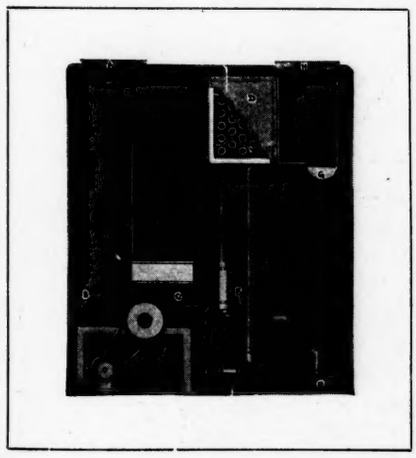
# NEW EQUIPMENT

In writing to HEATING & VENTILATING for additional information mention the H. & V. Item number



## Dunham Heating System

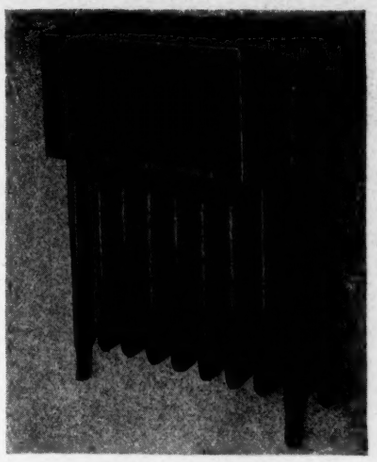
NAME—Dunham Sav-T-heat system.
PURPOSE — For heating buildings using gas as a fuel.
FEATURES — This heating system is unusual in that the products of combustion from the gas burner are mixed with water vapor and then passed through the radiators. The steam in this mixture condenses and flows back to the boiler while the flue gases are exhausted through a stack to the outside. This system is available either in the form of a warm air system or radiator system. With either the operating principle is the same. Gas is burned in the combustion chamber of the generator and is then drawn into the mixing chamber which contains water. Here the hot gas and water form a gas vapor combination. This mixture leaves the generator at a temperature slightly below 212F and is drawn into the supply main and then to the radiators. In the radiators the heat of the mixture is extracted and distributed into the room. The gas is cooled and the water vapor condenses. The condensate flows by gravity back into the water compartment where the excess water overflows to the sewer while the non-condensable gases are discharged into the copper vent tube of the system and thence to the atmosphere above the roof line. A special exhaust fan maintains a slight vacuum on the return from the radiators so that there is a positive draft on the burner at all times and so that the mixture will flow through the radiators. The manufacturer claims that a high combustion efficiency is obtained and because no chimney is used there is no danger of downdrafts decreasing the efficiency. It is said that this system operates with from 90 to 95% efficiency in actual field use.
CAPACITY—100,000 B.t.u. per hr. input.
LITERATURE AVAILABLE — Form No. 301.
MADE BY—*C. A. Dunham Co., 450 E. Ohio St., Chicago.*
H. & V. ITEM NO. 38A



## American Resistance Gage

NAME—Sonistat.
PURPOSE—A warning device for indicating when air filters should be changed.

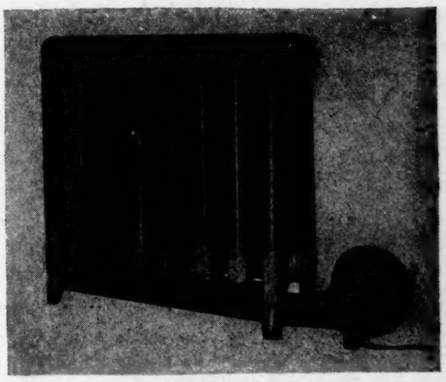
FEATURES—This device is made up of a specially calibrated metal reed which is designated to sound a warning note when the filter resistance reaches 0.25 in. w.g. This signal is continuous when fan is in operation and indicates to the operator of the system that the filters should be cleaned or changed. The device should be placed on the outside of the duct or housing between the filters and the fan intake so that it will be affected by the pressure differential caused by the resistance of the filter. This device is applicable to furnace blowers, unit air conditioners, and similar equipment in which the filter resistance should not exceed ¼ in. w.g.
MADE BY—*American Air Filter Co., Inc., Louisville, Ky.*
H. & V. ITEM NO. 38B



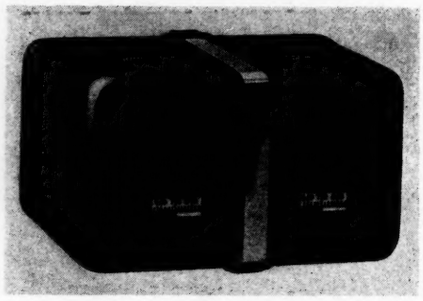
*Arco radiator booster 333*

## Arco Radiator Booster

NAME AND MODEL NUMBER—Arco radiator booster Nos. 333 and 334.
PURPOSE—For increasing the capacity of radiators.
FEATURES—The first model consists of two propeller-type fans mounted in an ivory enameled jacket and designed to draw air through the grille and discharge it through the radiator. Unit is attached to tube-type radiators by means of brackets which hook over the top of the radiator. A switch mounted on the side of the unit controls the operation. Model 334 is designed to fit under tube-type radiators which have a 3-in. clearance from the floor. It consists of a small motor-driven blower, the discharge of which is attached to a distribution tube. This tube is placed under the radiator and when the blower is in operation air is blown upward through the radiator. Either of these units is said to be capable of increasing the output of a radiator as much as 100%, depending on the size of the radiator.
MADE BY—*American Radiator Co., 40 W. 40th St., New York.*
H. & V. ITEM NO. 38C



*Arco radiator booster 334*

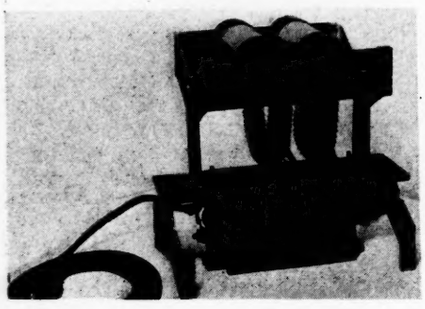


### Barber-Colman Thermostat

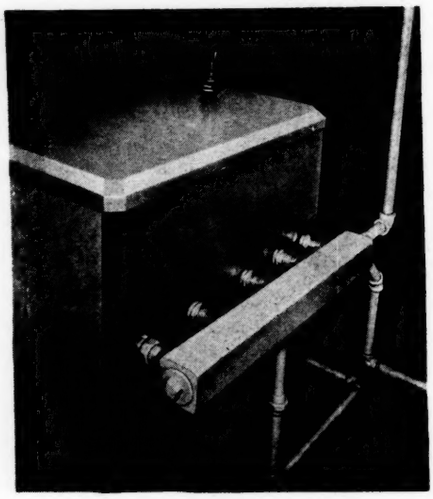
NAME—Supertherm.
PURPOSE—For controlling the operation of heating equipment to give a predetermined room temperature.
FEATURES—This thermostat is designed to be mounted on the wall with the long dimension horizontal rather than vertical. The manufacturer states that this horizontal design produces a thermostat with a rapid response. The bimetallic element is also mounted in a horizontal plane so as to present the largest surface of the element to the vertical movement of air which is most prevalent in the room. The two horizontal windows in the front of the instrument carry the adjusting scale and the thermometer. The liquid type thermometer lies horizontal in the left window and the adjusting scale is in the right window. The adjusting pointer is made of a slider of red bakelite, thus giving a current similar to the rising liquid in the thermometer at the left. This instrument may be used in two- or three-wire circuits and is available with or without heat anticipating features, with or without detents giving snap action or floating action. All instruments are equipped with locking shoes which allow the adjusting lever to be locked in any position or locked against a high or low limit position. The Supertherm is housed in a bakelite case with metal bands covering the mounting screws. Standard color is black bakelite with polished stainless steel band, but special colors are available on order.
MADE BY—*Barber-Colman Co., Rockford, Ill.*
H. & V. ITEM NO. 38D

### Lowell Duct Ionizer

NAME AND MODEL NUMBER—Lowell Model DT ionizer unit.

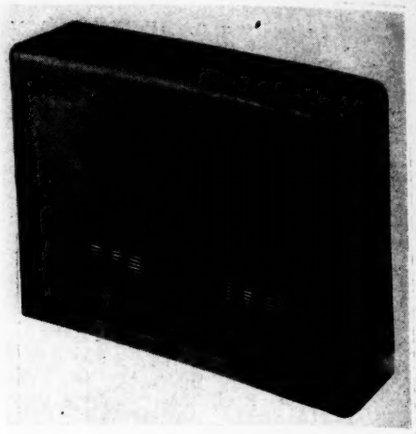


PURPOSE — To generate ozone and thus oxidize disagreeable odors and prevent bacteria growth.
FEATURES — The ionizing tubes are arranged in such a manner that by turning a dial a regulated amount of ionizing effect can be produced. A high voltage transformer of the special enclosed type provides the high potential electric current necessary for the ionizing effect. The device is constructed of aluminum, bakelite, and special composition ionizing tubes. It is designed to be placed in the fan chamber or return duct of any type of heating, ventilating, or air conditioning system. The manufacturer states that the ozone will prevent the staleness that is sometimes encountered in recirculating air systems.
MADE BY — *Lowell Air Conditioning Corp., Otis Bldg., Philadelphia, Pa.*
H. & V. ITEM NO. 38E



### Taco Water Heater

NAME—Taco flexible header heater.
PURPOSE—A combined boiler header and hot water heater for supplying domestic hot water.
FEATURES—In many cases the construction of the sectional cast iron boiler is such that when installing an indirect heater it is necessary to first provide for free circulation between the sections by means of a header connected to each section which in turn is connected to the heater. This is done to prevent the steaming in one section alone. This heater is so constructed that it may be used in place of the header, thus reducing installation and operating costs. Only the connection to one section is made rigid, the other sections are connected to the heater by means of flexible bellows.
SIZES AND CAPACITIES—Seven sizes with capacities ranging from 40 to 144 gal. per 3 hr., 100F rise with 212F boiler water.
MADE BY — *Taco Heaters, Inc., 342 Madison Ave., New York.*
H. & V. ITEM NO. 38F

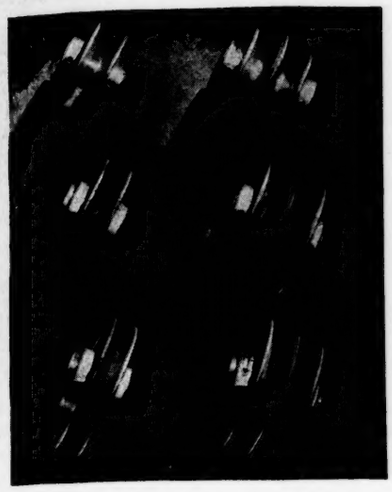


### Young Forced Convector

NAME AND MODEL NUMBER—Young forced convector type FC.
PURPOSE—For use with steam or hot water systems for heating entrance halls where unit heater operation is desired with convector appearance.
FEATURES—Consists of two multivane fans and one direct connected motor of either constant or two-speed type. Can be used with either steam or hot water systems. Has a brown crackle finish.
SIZES AND CAPACITIES—Three sizes, 27,000 B.t.u. per hr. in the small size; 72,000 B.t.u. per hr in the medium size, and 102,000 B.t.u. per hr. in the large size.
LITERATURE AVAILABLE Catalog 6538.
MADE BY—*Young Radiator Co., Racine, Wis.*
H. & V. ITEM NO. 38G

### Black & Decker Electric Drill

NAME—"Shorty" electric drill.
PURPOSE — For drilling in cramped quarters.
FEATURES—Chuck spindle is mounted at right angles to the motor and reduces the overall working length of the tool to 4½ in. Deep vents cast in the gear-head provide for free passage of air even though the tool is grasped by the head.
SIZES—3/16 and ¼ in. sizes.
MADE BY—*The Black & Decker Mfg. Co., Towson, Md.*
H. & V. ITEM NO. 38H



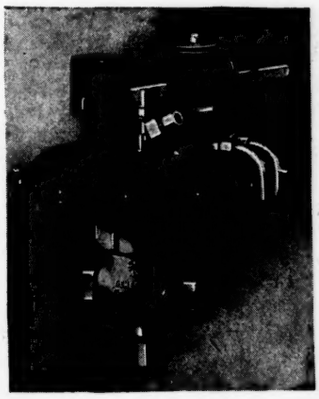
## Allis-Chalmers Adjustable Sheaves

NAME—Double groove adjustable pitch diameter texsteel sheaves.
PURPOSE—For providing a variable speed drive and speed variations up to 33% where the load does not exceed 3 hp.
FEATURES—The pitch diameters on these sheaves are changed by stopping the motor, relieving the belt tension, releasing the head locking screw, and turning the adjustable plate in or out to the desired pitch diameter.

Sheaves are furnished complete with a wrench for the head locking screw and a circular calculator showing directly the speeds of the driven unit. This company has also added an intermediate diameter texsteel sheave to its single groove adjustable diameter series. These also have a feed variation of 33% for ratings up to 1½ hp.
SIZES AVAILABLE—Three sizes of 2 to 3 in., 2½ to 3½, and 3½ to 4 in. variations, respectively.
MADE BY—*Allis-Chalmers Mfg. Co., Milwaukee, Wis.*
H. & V. ITEM NO. 38I

## General Controls Gas Valve

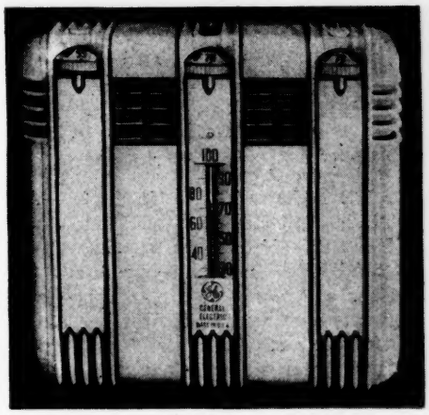
NAME AND MODEL NUMBER—General slow-opening gas valve, type B-55-1.
PURPOSE—A slow-opening gas valve for use with domestic conversion burners.
FEATURES—This valve is two-wire, electrically operated, and is said to not require oiling or servicing. Its function is the automatic control of gas flow and damper opening. Gas is bled into the furnace only during the opening period which gives a flare pilot effect to ignite the main gas flow. The operation of the damper arm is separate from the main valve stem. The main valve closes even when an external object may hold up the damper arm. Opening speed is adjustable from 5 to 60 sec. Fast shutoff is factory set. Union nut construction permits quick adjustment of damper arm into position.



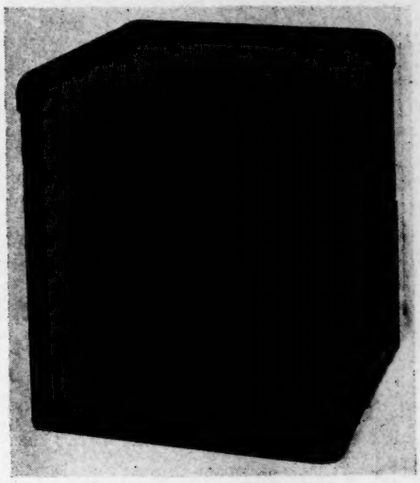
MADE BY—*General Controls Co., 450 E. Ohio St., Chicago.*
H. & V. ITEM NO. 38J

## G-E Humidistat and Thermostat

NAME AND MODEL NUMBER — G-E humidistat and night and day thermostat. Model 21ND1A1.
PURPOSE — For controlling humidity and temperature.
FEATURES—This device is composed of a humidistat and a night and day thermostat. The casing is made of ivory plastic with gold lettering and



trim. Humidistat is at the left while the two thermostats are at the middle and right. The thermostatic temperatures may be maintained or selected by a time switch which automatically changes from a specified day temperature to a specified night temperature and back again at a predetermined time. Humidistat consists of a multiple hair element, electrical contacts, and a magnet to cause snap-action. The dial is calibrated from 20% to 60% relative humidity range. The two-wire control blade opens on a relative humidity rise and closes on a relative humidity drop. Humidistat is used for low voltage in connection with a transformer. The thermostat incorporates preheat coils to reduce over-shooting of room temperatures. This humidistat is available in a separate case as are the thermostats. Thermostat and humidistat are also available in special combination.
MADE BY — *General Electric Co., Schenectady, N. Y.*
H. & V. ITEM NO. 38K

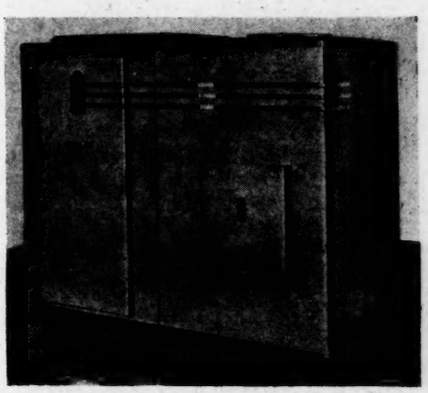


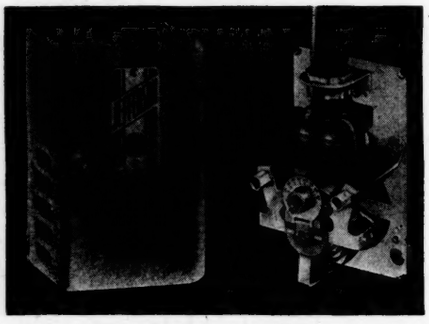
## Lau Furnace Blower

NAME AND MODEL NUMBER — Bloaire package unit furnace blower No. 401.
PURPOSE—For use with warm air heating systems to supply forced and filtered air.
FEATURES—Unit consists of a centrifugal fan, motor and filters. The motor is equipped with an automatic belt tightening device to provide proper tension at all times for maximum efficiency. An automatic cutout stops motor in case of overload. Two filters 15 x 20 in. are included as standard equipment. These filters are housed in leakproof filter frames.
CAPACITIES—Unit delivers approximately 1000 c.f.m.
MADE BY—*The Lau Blower Co., 954 E. Monument Ave., Dayton, Ohio.*
H. & V. ITEM NO. 38L

## Heil Winter Air Conditioning Unit

NAME — Heil winter air conditioning unit.
PURPOSE—For winter air conditioning.
FEATURES—The case on this unit has been completely redesigned. The new case is made with a hammered copper finish with a chromium trim. The furnace, burner, and blower are the same as in the older model.
MADE BY—*The Heil Co., 3000 W. Montana St., Milwaukee, Wis.*
H. & V. ITEM NO. 38M





### Jefferson Temperature Control

NAME—Concealed adjustment type of air switch control.
PURPOSE—For controlling air temperatures.
FEATURES—This control is of the concealed adjustment type for use in locations where easy adjustment of thermostats is not desirable. The control knob and scale plate are removed from the outside of the enclosing case and the range adjustment can only be made by removing the case. The absence of the usual control knob is said to remove the danger of tampering.
MADE BY—*Jefferson Electric Co., Bellwood, Ill.*
H. & V. ITEM NO. 38N

### Torrington Flexible Coupling

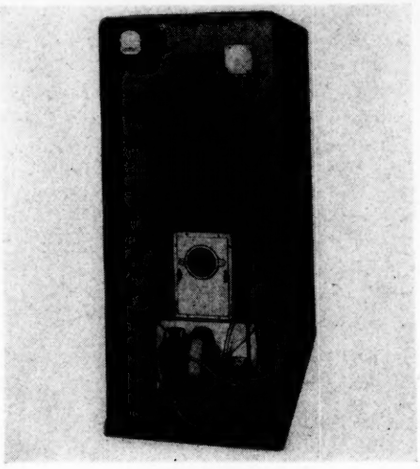
NAME—Torrington flexible coupling.
PURPOSE—A spline type flexible coupling for use in gun-type oil burners.
FEATURES—The sleeve end of the coupling is a sliding fit on the wheel hub and when assembled covers the hub set screw, preventing this screw from working out. Power is transmitted from the blower wheel by four splined grooves in the hub to four splined ribs in the sleeve of the coupling. The chief advantages of this coupling are said to be maximum flexibility with minimum of parts; simple assembly in burner with only one set screw; permits the use of cheaper and shorter shaft motors; eliminates end thrust on oil pump and resulting leaks.
MADE BY—*The Torrington Mfg. Co., 28 Franklin St., Torrington, Conn.*
H. & V. ITEM NO. 38O

### Armco Zincgrip

NAME—Armco Zincgrip.
PURPOSE—A heavily zinc coated steel sheet for use in severe drawing or forming operations.
FEATURES—The manufacturer states that Armco Zincgrip will not crack or peel when subjected to severe drawing or forming operations. It is provided in either coils or sheets and carries from 50 or 75% more protective zinc than tight-coat sheets generally used for fabricated products.
MADE BY—*The American Rolling Mill Co., Middletown, Ohio.*
H. & V. ITEM NO. 38P

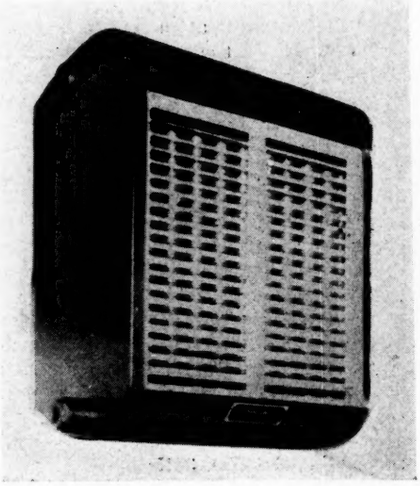
### U.S. Winter Air Conditioning Unit

NAME—Capitolaire conditioning unit.
PURPOSE—For winter air conditioning using oil as a fuel.
FEATURES—This winter air conditioning unit is furnished complete with oil burner, room thermostat and automatic humidifier assembled in one crated unit ready for chimney connection. Each unit has been under fire and electrically tested before shipment is made.
SIZE—100,000 B.t.u. per hr. at register.
MADE BY—*U. S. Radiator Corp., 44 Cadillac Sq., Detroit, Mich.*
H. & V. ITEM NO. 38Q



### Airtherm Airblanket Unit Heater

NAME AND MODEL NUMBER—Airblanket unit heater, Model TAB.
PURPOSE—For space heating.
FEATURES—The unit produces two separate streams of air, a top one of unheated or relatively cold air and a bottom one of warm air. The purpose of the upper stream is to prevent the rise of the lower stream of warmed air, thereby producing an even distribution of temperature in the room.
SIZES—Twenty.
MADE BY—*Airtherm Manufacturing Co., 1474 S. Vandeventer, St. Louis, Mo.*
H. & V. ITEM NO. 38R

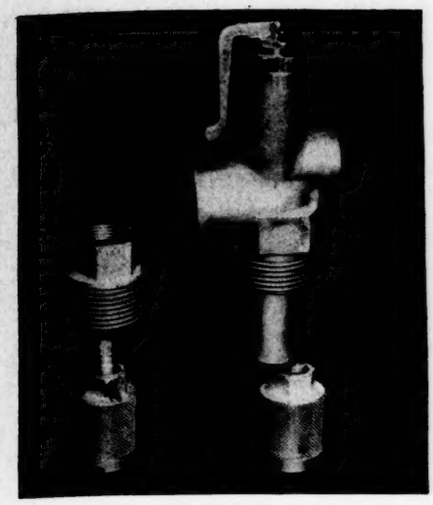


### Boiler Room Smoke Control

NAME AND MODEL NUMBER—Sentry smoke control No. 2M (a.c. or d.c.).
PURPOSE—A smoke alarm and combustion guide.
FEATURES—This device is said to act not only as a smoke signal but to furnish means of keeping a constant check on the efficiency of fires. The device reports through an audible signal when the stack gases are too thick or smoking, as well as when the gases are too thin due to an excess of air. As the smoke gets thicker or the excess air increases, the device sounds an alarm faster and faster. In addition to this a large dial, which reads 0 when the combustion is most efficient, moves when a change in combustion takes place.
MADE BY—*Boiler Room Equipment, Inc., 45 W. 45th St., New York.*
H. & V. ITEM NO. 38S

### Whitney-Jensen Rolling Machine

NAME AND MODEL NUMBER—Whitney-Jensen duplex rolling machine No. 75.
PURPOSE—For making air conditioning and ventilating ducts and other similar forms using folded seams such as the Pittsburgh or hammer lock.
FEATURES—This machine has a double set of rolls which permit one set to be used for forming corner locks and the other for drive cleats. Either or both sets of rolls may be used at any time so that no changeover of the machine is required when switching from one type of work to the other. The machine will handle material up to 18 gage and feeds the sheets through the rolls at a speed of 38 f.p.m. It is said that the cost of making ductwork with folded seams can be reduced by 50% or more with this machine. With special rolls, many other forms of folds and seams can be made.
MADE BY—*Whitney Metal Tool Co., Rockford, Ill.*
H. & V. ITEM NO. 38T

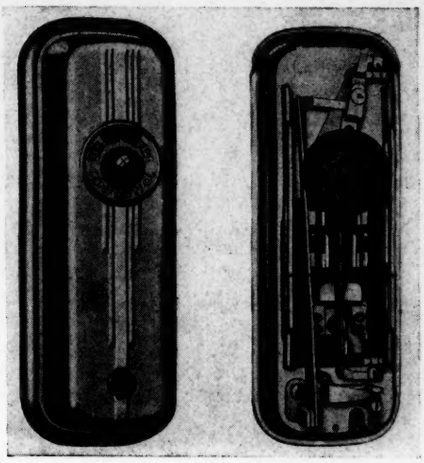


## Spencer Relief Valves

NAME AND MODEL NUMBER—Left, pressure relief valve No. ITM; right, combination pressure and relief valve No. 2TMH.
PURPOSE — For relieving excessive temperature or pressure conditions in the storage tanks of hot water heating systems.
FEATURES—Valves are designed to be immersed directly in the water in the storage tank. The thermal element employed for temperature relief is a snap-acting thermostatic disc. Relief valves and valve seats are protected from dirt particles by a fine mesh screening. In the 2TMH type the water outlet from the tank is incorporated in the casting.
SIZES AND CAPACITIES — 75,000 B.t.u. per hr. input and designed to fit 1 in. I.P.S. tappings.
MADE BY—*Spencer Thermostat Co., 34 Forest St., Attleboro, Mass.*
H. & V. ITEM NO. 38U

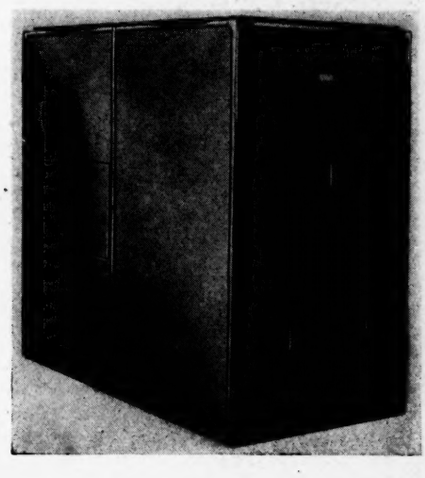
## Friez Effective Temperature Controller

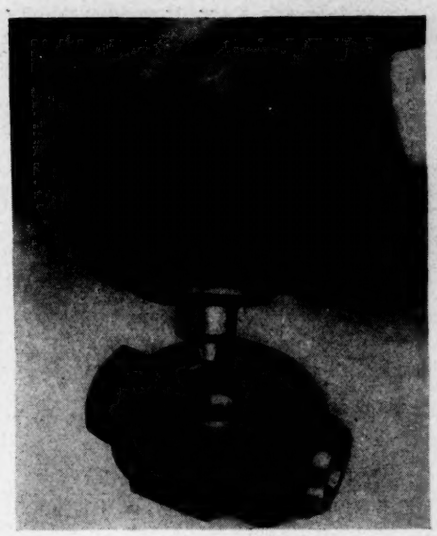
NAME—Friez Comfortrol.
PURPOSE—An instrument which automatically varies its own dry bulb point of control to compensate for the uncontrolled variations of humidity which exist in simple heating and cooling applications.
FEATURES—This instrument is really a humidity set thermostat, that is, the point of the dry bulb control is varied automatically to suit the relative humidity existing at the moment. When the humidity rises the dry bulb point of control is automatically lowered so as to maintain the same effective temperature. This is achieved by a bimetallic temperature element and a multiple hair element. The moving end of the bimetallic element is connected to a moving contact blade through a linkage movement and this in turn to the humidity sensitive human hair element. This device can be obtained either for maintaining a single temperature or with the day-night feature. Standard effective temperature range 56 to 79F.
MADE BY—*Julien P. Friez & Sons, Inc., Baltimore, Md.*
H. & V. ITEM NO. 38V



## Duo-Therm Winter Air Conditioning Unit

NAME — Duo-Therm winter air conditioning unit.
PURPOSE—For winter air conditioning using oil as a fuel.
FEATURES—Unit consists of a special preheater, filters, centrifugal blower and oil-fired furnace. Air entering the unit passes through filters, a humidifier, and over a special waste stopper in the rear compartment. This is a heating surface through which the flue gases pass before going to the chimney. Special features include a double casing for minimizing heat loss, three-way stack outlets, built-in draft regulator, automatic humidifier, noise insulation, and a patented dual-chamber burner.
SIZES AND CAPACITIES—Three sizes with maximum B.t.u. output from 51,500 to 112,000.
MADE BY — *Duo-Therm Division, Motor Wheel Corp., Lansing, Mich.*
H. & V. ITEM NO. 38W



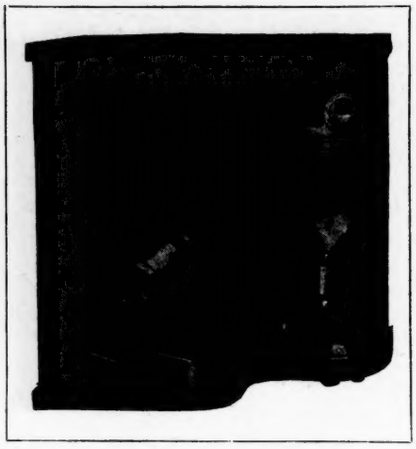


## Hancock Valve

NAME—Hancock Duravalve.
PURPOSE—For high pressure and high temperature service (1500 lb. at 950F).
FEATURES—The manufacturers claim that these valves with their internal Stellite seats put an end to valve maintenance due to steam leaking between the valve seat ring and the body. Valves are made in one basic size and then tapped or bored for welding for ½-, ¾- or 1-in. pipe.
MADE BY — *Hancock Valve Division, Manning, Maxwell & Moore, Inc., Bridgeport, Conn.*
H. & V. ITEM NO. 38X

## Dunham Year-Round Conditioning Unit

NAME—Dunham Temperator.
PURPOSE—For complete air conditioning including warming and humidification for the heating season and cooling and dehumidification for the summer, with filtering, ventilation and circulation the year-round.
FEATURES — The unit consists of a mixing damper, filters, two centrifugal blowers, a humidifying device and combined heating and cooling coil. This heating and cooling coil is so designed that there is three times greater capacity in cooling surface than is used for heating. Three fan speeds add to the adjustment for heating and cooling a given space. A unique damper either automatically controlled or hand-operated is used to regulate the amount of outside and inside air to be mixed and circulated. Changing from a heating to a cooling unit requires only the turning of a valve to provide cold water instead of steam.
SIZES AND CAPACITIES — Made in three sizes ranging from 100 to 350 sq. ft. of equivalent radiator surface. Cooling capacity when using 40F water is 9000 B.t.u. per hr.
MADE BY—*C. A. Dunham Co., 450 E. Ohio St., Chicago.*
H. & V. ITEM NO. 38Y



### Arco Thermo Humidifier

NAME AND MODEL NUMBER—Arco thermo humidifier No. 7000.
PURPOSE—For supplying up to 4 gal. of moisture to the air per day.
FEATURES—This device is made of heavy brass and is mounted with a dustproof electric motor and blower. The air is transmitted by the blower through the humidifier which contains a self-cleaning vaporizing nozzle. From the humidifier the air is carried by means of ducts to an outlet grille in a central room of the residence. A water supply line is connected to the humidifier and the flow of water is controlled by an electric valve. This valve can either be controlled by room humidistat or by a manually-operated switch. An Arco Thermo humidifier interrupter can be supplied with this device to prevent over-humidification in cold weather. This control has been adjusted so that the humidifier is automatically shut off when the outside temperature drops below 15F.
MADE BY—*American Radiator Co., 40 W. 40th St., New York.*
H. & V. ITEM NO. 38Z

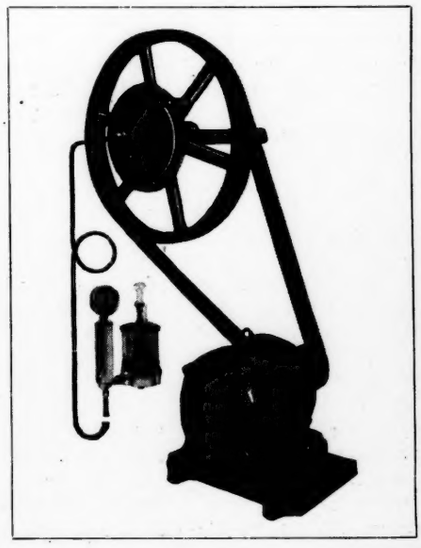
### Illinois Heating Control System

NAME—Illinois Selective Pressure control system.
PURPOSE—To automatically control the operation of vapor or vacuum heating systems by regulating the pressures maintained and the combustion rate.
FEATURES—This control system consists essentially of a device for regulating the combustion rate such as a Vapor-stat or thermostat, a Selective Controller for controlling the operation of the vacuum pump in accordance with the return line temperature and pressure and a Guide Chart to show the boiler pressure to be maintained for various weather conditions. The Selective Controller automatically operates the vacuum pump to maintain the maximum vacuum possible on the return lines for the temperature existing at the pump. When this is done the maximum differential is maintained between the supply and return main pressures which is said to cause the rapid circulation of steam throughout the entire system to give even heating conditions.

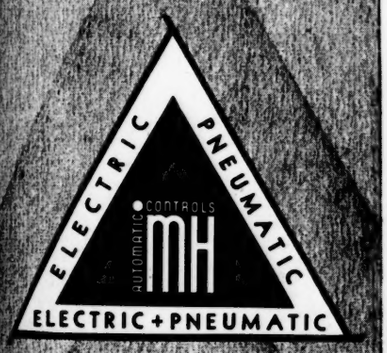
METHOD OF OPERATION—When the heating system is started in the morning both the firing device (oil burner, stoker, blower or like device) and the vacuum pump are put into operation. The vacuum pump quickly drops the pressure in the entire system, including the boiler, to a high vacuum. The returns are cold and the pump continues to operate until the maximum cut-out vacuum is reached when it stops, for example, at 25 in. of vacuum. When the return pressure increases, for instance to 21 in. of vacuum, the pump starts again. Meanwhile, the boiler pressure is rising rapidly and this continues until the boiler radiator pressure reaches the selected boiler pressure, say 1 lb., to take care of morning pickup. Inasmuch as the returns are relatively cool during this heating-up period, there will exist the maximum pressure differential between the two sides of the system. The temperature in the returns gradually rises and the Selective Controller responds to this rise causing the pump to operate between progressively rising pressure cut-in and cut-out points. This continues until the temperature in the returns is stabilized, when the pump will operate for instance from 6 to 10 in. of vacuum for this supply pressure condition. When the building reaches the desired temperature the Vapor-stat is reset for, say, 16 in. vacuum and the blower ceases to operate until the boiler pressure drops to this newly selected level. The returns will gradually cool off as the pressure in the radiators falls and the Selective Controller responds by operating the pump between gradually lowering pressure cut-in and cut-out points until the temperature stabilizes for the new selected boiler pressure and the pump operates from say 21 to 25 in. of vacuum. When weather conditions are mild the Vapor-stat can be turned to the off position so that the blower will not operate at all. The boiler pressure will then drop to the lowest pressure level established by the Selective Controller and under this condition return temperature will probably not be in excess of 70 to 80F. When the pump starts at 21 in. vacuum, the temperature in the boiler will be approximately 157F and as the pump drops the pressure to 25 in. of vacuum where the boiling point is approximately 133F, the sensible heat stored in the water of the boiler will be converted into latent heat of evaporation. This will cause shots of steam to be supplied periodically to all radiators. The radiators will then cool, as the pressure within the system increases to the cut-in pressure and the water in the boiler again stores up heat from the slowly burning fuel until the temperature again reaches 157F.
SIZES — For heating systems of all types of buildings, such as large residences, schools, office buildings, and factories.
MADE BY — *Illinois Engineering Co., Racine Ave. at 21st St., Chicago.*
H. & V. ITEM NO. 38AA

### U.S. Airco Variable Speed Pulley

NAME—Variable speed pulley and hydraulic remote control.
PURPOSE—For remote control of the speed of blowers.
FEATURES—This device is a friction type of clutch controlled by the pressure carried on a diaphragm contained in the pressure unit which is screwed on to the end of the blower shaft. The pressure is varied by the operator by the manipulation of the pump unit which can be remotely located at any convenient point. The blower pulley is not keyed to the impeller shaft but turns free on a tapered roller bearing and is driven at a constant speed by the motor. Attached to and concentric with the blower pulley is a driving disc which also rotates at constant speed with the blower pulley. Riveted to each side of the driving disc is a special asbestos friction disc of the correct area to transmit the load required at full speed. This driving disc rotates between a back and front pressure plate each of which is keyed to the blower shaft. As pressure is applied on the diaphragm of the pressure unit the pressure plates are forced against the friction surfaces of the driving disc and the speed of the blower is increased. It is possible to obtain a range of speeds from idling to maximum pulley ratio.
SIZES—For blowers requiring from 3 to 30 hp.
MADE BY—*United States Air Conditioning Corp., 2139 Kennedy St., N.E., Minneapolis, Minn.*
H. & V. ITEM NO. 38BB



# THE MODUTROL SYSTEM IS FLEXIBLE ENOUGH TO MEET *any* CONTROL PROBLEM!




ELECTRIC ELECTRIC+PNEUMATIC PNEUMATIC

**TO meet any control problem it is necessary that Automatic Controls not only satisfy the temperature and air conditioning requirements, but that operating costs be held to a minimum. The Minneapolis-Honeywell Modutrol System is flexible enough in its application and accurate and dependable enough in its performance to exactly meet the requirements of any building, large or small, old or new. It is the only system which includes Electric, Pneumatic or a combination of Electric and Pneumatic Controls. For the best possible results at the lowest operating costs, recommend and install the Minneapolis-Honeywell Modutrol System. A Minneapolis-Honeywell engineer is available to assist you with any control problem. Minneapolis-Honeywell Regulator Co., 2715 Fourth Ave. So., Minneapolis, Minn. Branch and distributing offices in all principal cities.**

■

***Dependable Controls Cost Less Than Service***

# MINNEAPOLIS-HONEYWELL

BROWN INDUSTRIAL INSTRUMENTS
NATIONAL PNEUMATIC CONTROLS

AUTOMATIC CONTROLS MH *Control Systems*

# Degree-Day Figures for January, 1938

**HEATING & VENTILATING continues its tenth year of publishing degree-day data for various large cities**

| | Albany, N. Y. | Atlanta, Ga. | Baltimore, Md. | Birmingham, Ala. | Boston, Mass. | Buffalo, N. Y. | Burlington, Vt. |
|---|---|---|---|---|---|---|---|
| Degree-days for January, 1938 ..... | 1313 | 663 | 896 | 596 | 1150 | 1261 | 1481 |
| Degree-days, Sept. 1, 1937 to Jan. 31, '38 | 3736 | 2071 | 2600 | 1841 | 3342 | 3887 | 4411 |
| Degree-days, Sept. 1, 1936 to Jan. 31, '37 | 3362 | 1488 | 2228 | 1279 | 3015 | 3424 | 4137 |
| Degree-days, Sept. 1, to Jan. 31, Normal | 3726 | 1813 | 2619 | 1502 | 3280 | 3612 | 4202 |

| | Cheyenne, Wyo. | Chicago, Ill. | Cincinnati, Ohio | Cleveland, Ohio | Columbus, Ohio | Denver, Colo. | Des Moines, Iowa |
|---|---|---|---|---|---|---|---|
| Degree-days for January, 1938 ..... | 1165 | 1230 | 1010 | 1118 | 1047 | 935 | 1313 |
| Degree-days, Sept. 1, 1937 to Jan. 31, '38 | 3950 | 3806 | 3218 | 3515 | 3368 | 3049 | 3976 |
| Degree-days, Sept. 1, 1936 to Jan. 31, '37 | 4425 | 3389 | 2748 | 2999 | 2872 | 3653 | 3943 |
| Degree-days, Sept. 1, to Jan. 31, Normal | 4104 | 3487 | 2795 | 3366 | 3133 | 3355 | 3762 |

| | Detroit, Mich. | Dodge City, Kan. | Duluth, Minn. | El Paso, Tex. | Erie, Pa. | Evansville, Ind. | Fort Wayne, Ind. |
|---|---|---|---|---|---|---|---|
| Degree-days for January, 1938 ..... | 1262 | 906 | 1747 | 557 | 1182 | 940 | 1201 |
| Degree-days, Sept. 1, 1937 to Jan. 31, '38 | 3953 | 2942 | 5482 | 1443 | 3672 | 2854 | 3850 |
| Degree-days, Sept. 1, 1936 to Jan. 31, '37 | 3526 | 3296 | 5661 | 1848 | 3128 | 2515 | 3370 |
| Degree-days, Sept. 1, to Jan. 31, Normal | 3584 | 3068 | 5161 | 1667 | 3336 | 2494 | 3405 |

| | Fort Worth, Tex. | Grand Rapids Mich. | Green Bay, Wis. | Harrisburg, Pa. | Hartford, Conn. | Indianapolis, Ind. | Ithaca, N. Y. |
|---|---|---|---|---|---|---|---|
| Degree-days for January, 1938 ..... | 495 | 1277 | 1536 | 1057 | 1189 | 1102 | 1235 |
| Degree-days, Sept. 1, 1937 to Jan. 31, '38 | 1553 | 3953 | 4653 | 3196 | 3482 | 3481 | 3850 |
| Degree-days, Sept. 1, 1936 to Jan. 31, '37 | 1761 | 3589 | 4466 | 2818 | 3088 | 3019 | 3514 |
| Degree-days, Sept. 1, to Jan. 31, Normal | 1406 | 3659 | 4420 | 3069 | 3316 | 3130 | 3706 |

| | Kansas City, Mo. | Knoxville, Tenn. | La Crosse, Wis. | Lansing, Mich. | Lincoln, Neb. | Little Rock, Ark. | Los Angeles, Calif. |
|---|---|---|---|---|---|---|---|
| Degree-days for January, 1938 ..... | 1021 | 763 | 1459 | 1320 | 1168 | 679 | 128 |
| Degree-days, Sept. 1, 1937 to Jan. 31, '38 | 3151 | 2451 | 4491 | 4220 | 3530 | 2053 | 368 |
| Degree-days, Sept. 1, 1936 to Jan. 31, '37 | 3117 | 1725 | 4525 | 3876 | 3821 | 1929 | 800 |
| Degree-days, Sept. 1, to Jan. 31, Normal | 2903 | 2298 | 4313 | 4001 | 3512 | 1798 | 750 |

| | Louisville, Ky. | Madison, Wis. | Memphis, Tenn. | Milwaukee, Wis. | Minneapolis, Minn. | Nashville, Tenn. | New Haven, Conn. |
|---|---|---|---|---|---|---|---|
| Degree-days for January, 1938 ..... | 953 | 1463 | 715 | 1338 | 1593 | 807 | 1118 |
| Degree-days, Sept. 1, 1937 to Jan. 31, '38 | 2944 | 4486 | 2111 | 4106 | 4784 | 2459 | 3258 |
| Degree-days, Sept. 1, 1936 to Jan. 31, '37 | 2511 | 4277 | 1872 | 3795 | 4878 | 1962 | 2897 |
| Degree-days, Sept. 1, to Jan. 31, Normal | 2526 | 4256 | 1871 | 3984 | 4550 | 2175 | 3219 |

| | New Orleans, La. | New York, N. Y. | Norfolk, Va. | Oklahoma City, Okla. | Omaha, Neb. | Peoria, Ill. | Philadelphia, Pa. |
|---|---|---|---|---|---|---|---|
| Degree-days for January, 1938 ..... | 336 | 1022 | 741 | 745 | 1255 | 1205 | 974 |
| Degree-days, Sept. 1, 1937 to Jan. 31, '38 | 953 | 2938 | 2094 | 2360 | 3759 | 3785 | 2830 |
| Degree-days, Sept. 1, 1936 to Jan. 31, '37 | 568 | 2542 | 1544 | 2518 | 4068 | 3468 | 2423 |
| Degree-days, Sept. 1, to Jan. 31, Normal | 735 | 2909 | 1933 | 2244 | 3639 | 3606 | 2736 |

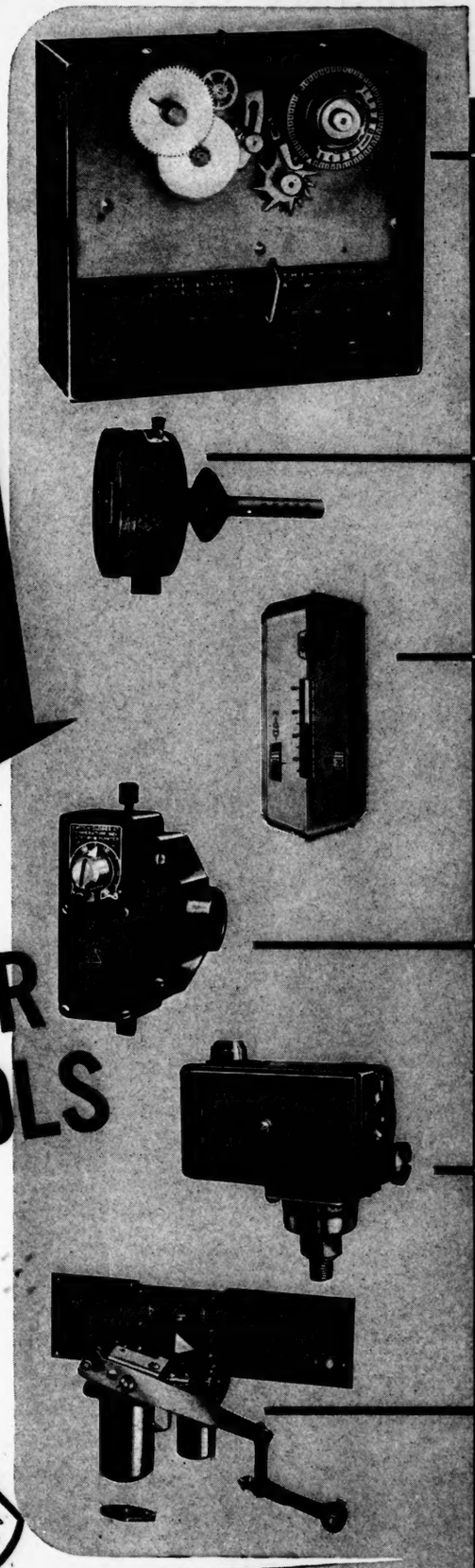
| | Pittsburgh, Pa. | Portland, Me. | Portland, Oreg. | Providence, R. I. | Reading, Pa. | Reno, Nev. | Richmond, Va. |
|---|---|---|---|---|---|---|---|
| Degree-days for January, 1938 ..... | 1013 | 1282 | 730 | 1131 | 1018 | 887 | 828 |
| Degree-days, Sept. 1, 1937 to Jan. 31, '38 | 3254 | 3859 | 2115 | 3340 | 3028 | 2767 | 2479 |
| Degree-days, Sept. 1, 1936 to Jan. 31, '37 | 2663 | 3608 | 2631 | 2980 | 2675 | 3684 | 2002 |
| Degree-days, Sept. 1, to Jan. 31, Normal | 3003 | 3817 | 2530 | 3246 | 3102 | 3326 | 2232 |

| | Rochester, N. Y. | St. Louis, Mo. | Salt Lake City, Utah | San Francisco, Calif. | Scranton, Pa. | Seattle, Wash. | Spokane, Wash. |
|---|---|---|---|---|---|---|---|
| Degree-days for January, 1938 ..... | 1250 | 976 | 873 | 419 | 1154 | 699 | 1038 |
| Degree-days, Sept. 1, 1937 to Jan. 31, '38 | 3844 | 3026 | 2605 | 1167 | 3576 | 2201 | 3258 |
| Degree-days, Sept. 1, 1936 to Jan. 31, '37 | 3350 | 2766 | 3699 | 1508 | 3170 | 2696 | 4268 |
| Degree-days, Sept. 1, to Jan. 31, Normal | 3666 | 2798 | 3259 | 1414 | 3446 | 2656 | 3755 |

| | Springfield, Ill. | Syracuse, N. Y. | Toledo, Ohio | Trenton, N. J. | Utica, N. Y. | Washington, D. C. | Wichita, Kan. |
|---|---|---|---|---|---|---|---|
| Degree-days for January, 1938 ..... | 1110 | 1241 | 1187 | 1018 | 1394 | 905 | 910 |
| Degree-days, Sept. 1, 1937 to Jan. 31, '38 | 3446 | 3726 | 3763 | 3044 | 4075 | 2712 | 2855 |
| Degree-days, Sept. 1, 1936 to Jan. 31, '37 | 3153 | 3327 | 3332 | 2676 | 3527 | 2273 | 2995 |
| Degree-days, Sept. 1, to Jan. 31, Normal | 3183 | 3768 | 3392 | 2774 | 3785 | 2711 | 2877 |

**Methods of applying the degree-day in connection with the operation of heating plants or for the prediction of fuel consumption of heating plants are described fully in the DEGREE-DAY HANDBOOK, published in 1937, by HEATING & VENTILATING, 148 Lafayette Street, New York, price $3. Degree-days as given above for a "normal" month or season are based on averages for a long period of years, ending about 1922. Averages covering different periods will disagree with the above figures slightly.**

# YOU PROFIT BY GOOD STOKER CONTROLS



### THE No. 387 "GENUINE DETROIT" STOKER TIME RELAY

The No. 387 Timer combines all the features and faculties of the well known "Genuine Detroit" Timer together with a time switch mechanism for day and night thermostat control. Settings are easily and quickly made. Where Stoker "Stand by" protection only is wanted, other "Genuine Detroit" Timers are available.

Write for Bulletin No. 94 covering a complete line of Timers and Time Relays, with and without Time Switches, Fused Safety Switches, etc.

### THE No. 255 FURNACE LIMIT SWITCH

Installed in the furnace bonnet, it breaks the stoker motor circuit at a given maximum allowable furnace temperature. Write for Bulletin No. 64.

### THE TWO-ELEVEN DAY AND NIGHT THERMOSTAT

This is an attractively styled instrument incorporating more advantages than any other similar unit available. It is sensitive, accurate and reliable. This model for use with No. 387 Time Relay (above). Also available in plain type (without day and night feature). Furnished with or without heat compensation (preheater). Write for Bulletin No. 86.

### THE No. 251 CLAMP-ON LIMIT SWITCH

Clamped to hot water pipe, the No. 251 limits stoker operation according to the temperature of the hot water. Refer to Bulletin No. 165.

### THE No. 250 MA-1 PRESSURE LIMIT CONTROL

Installed at the boiler, it breaks the stoker motor circuit at a predetermined point, thus preventing excess steam pressures. Quickly and easily adjusted.

### THE No. 431 DUCT DAMPER MOTOR

The No. 431 provides the means for zone control of forced warm air systems. It is quiet in operation, is easily installed, needs no attention in service, and is exceptionally durable. Write for Bulletin No. 66.

*Reg. U. S. Patent Office

**DETROIT LUBRICATOR COMPANY**

DETROIT, MICHIGAN, U. S. A. • 5900 TRUMBULL AVE.

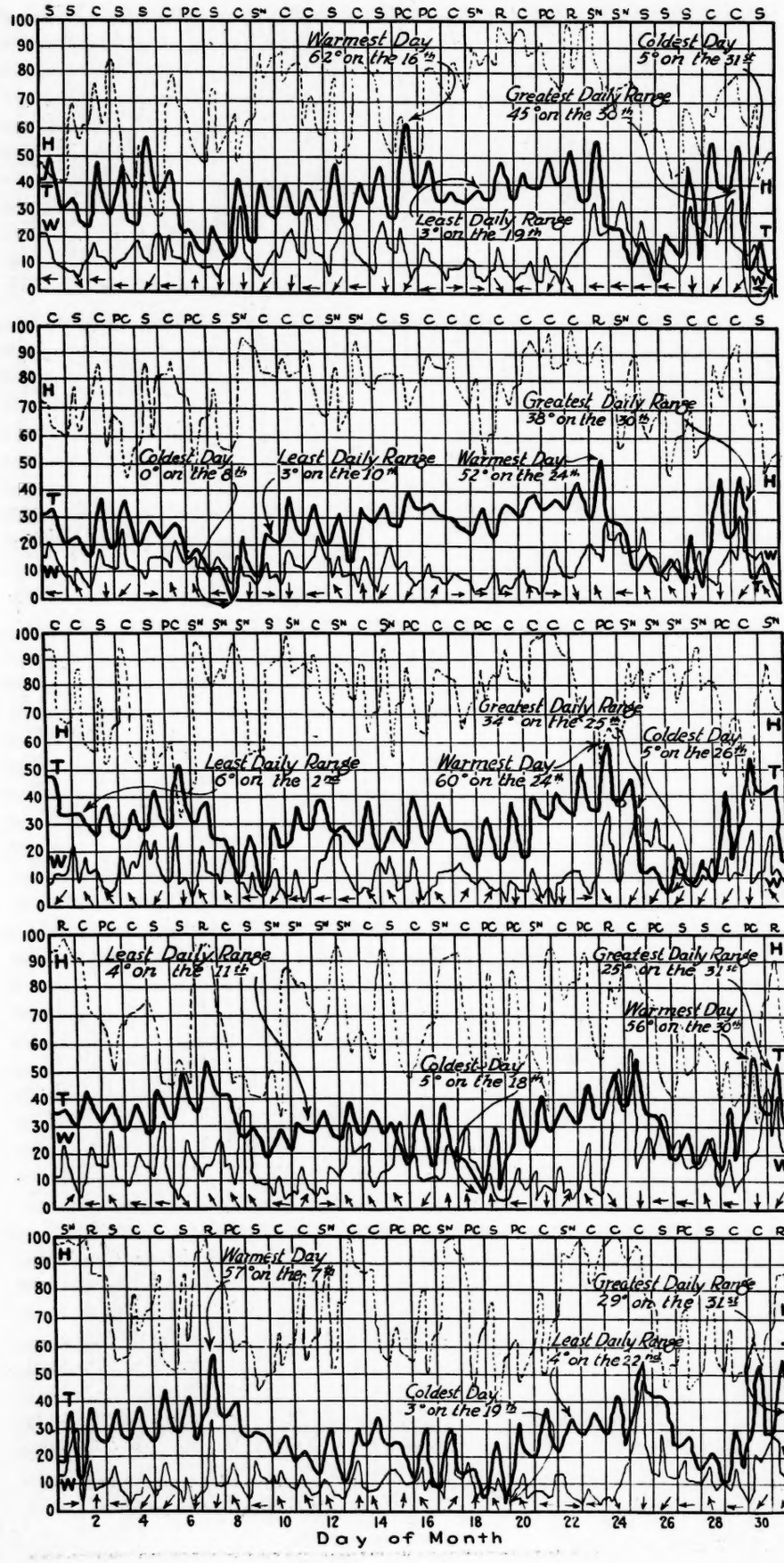
NEW YORK, N. Y.—40 WEST 40th ST. • CHICAGO, ILL.—816 S. Michigan Ave.

DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

Canadian Representative—RAILWAY AND ENGINEERING SPECIALTIES LIMITED, Montreal, Toronto, Winnipeg

# THE WEATHER FOR JANUARY, 1938

Plotted from records compiled for HEATING & VENTILATING by the U. S. Weather Bureau. Heavy curves (T), dry bulb temperatures in deg. F.; dotted lines (H), per cent relative humidity from readings at 8 a.m., noon, and 8 p.m. Light lines (W), wind velocity in m.p.h. Arrows indicate wind direction at maximum velocity, north being arrow pointing up, etc.; S—clear; PC—partly cloudy; C—cloudy; R—rain; Sn—snow.


Warmest Day 62° on the 16th
Coldest Day 5° on the 31st
Greatest Daily Range 45° on the 30th
Least Daily Range 3° on the 19th
Coldest Day 0° on the 8th
Least Daily Range 3° on the 10th
Warmest Day 52° on the 24th
Greatest Daily Range 38° on the 30th
Greatest Daily Range 34° on the 25th
Coldest Day 5° on the 26th
Least Daily Range 6° on the 2nd
Warmest Day 60° on the 24th
Least Daily Range 4° on the 11th
Greatest Daily Range 25° on the 31st
Warmest Day 56° on the 30th
Coldest Day 5° on the 18th
Warmest Day 57° on the 7th
Greatest Daily Range 29° on the 31st
Least Daily Range 4° on the 22nd
Coldest Day 3° on the 19th
H
T
W
2 4 6 8 10 12 14 16 18 20 22 24 26 28 30
Day of Month

## St. Louis

Mean temp. for month, 33.4F; aver. wind velocity, 13.4 m.p.h.; prevailing direction of wind, W.

## Chicago

Mean temp. for month, 25.4F; aver. wind velocity, 10.9 m.p.h.; prevailing direction of wind, W.

## Pittsburgh

Mean temp. for month, 29.6F; aver. wind velocity, 11.1 m.p.h.; prevailing direction of wind, SW.

## New York

Mean temp. for month, 32.0F; aver. wind velocity, 14.3 m.p.h.; prevailing direction of wind, N.W.

## Boston

Mean temp. for month, 28.0F; aver. wind velocity, 10.7 m.p.h.; prevailing direction of wind, W.

ADSCO INTERNALLY-EXTERNALLY GUIDED JOINT

**Buy This Joint for DEPENDABILITY**

A completely guided joint—inside and out. Slip cannot pull out of body. No metal to metal contact between body and polished slip surface. Available in semi-steel, cast steel, or wrought steel for high or low pressures in single or double joints with or without anchor base and service connections.

*Write for Bulletin No. 35-20 V*

AMERICAN DISTRICT STEAM COMPANY
NORTH TONAWANDA, N.Y.
IN BUSINESS OVER SIXTY YEARS

# TESTED AND PROVEN

No Master product is ever made available until it has been thoroughly tested and proven to be worthy to bear the Master name. The model B-22 has proven its dependability not only in our exhaustive tests, but in use under actual operating conditions in the many years it has been sold and installed all over the country.

**TYPE B-22**

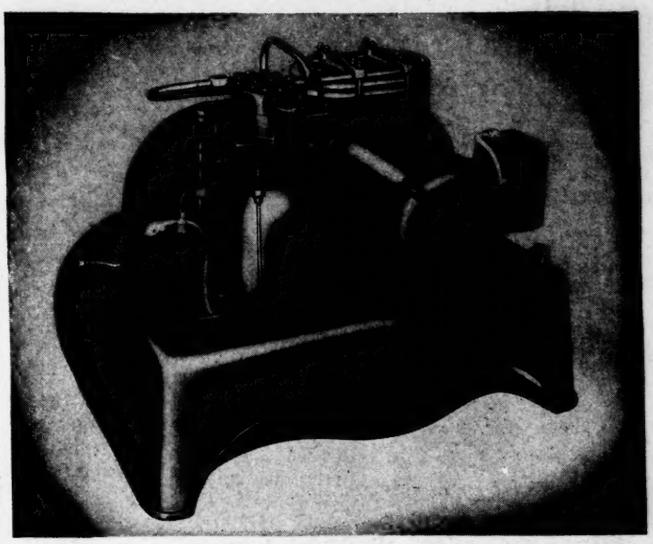
A two position heat regulator that is ideal for the small home. It is sturdily made and dependability and precision are built right into it. A noiseless device that will outlast the heating plant.

Made by the makers of the famous Type B-144, the original gradual control heat regulator.

**WHITE MFG. CO.**
2364 University Ave.
ST. PAUL, MINN.

# MASTER HEAT REGULATOR

# SUMMER COOLING
## *is a* GROWING *business*



## *safeguard* SALES *and* PROFITS *with* UNIVERSAL COOLER *units*

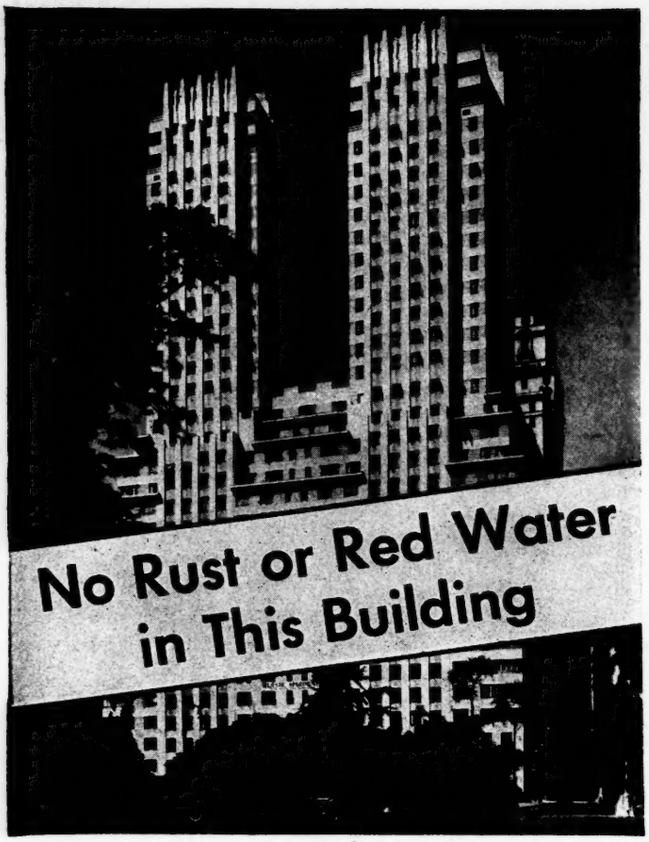
*A tabulation of the residential cooling installations in 55 cities shows that of the 4,000 jobs installed to date, 1,500 or 37½ per cent were sold last year. Conservative estimates place 1938 sales at 3,000 jobs . . . an increase of 75 per cent over the total number now in use.*

The rising trend of Summer Cooling points the way to new profits for manufacturers of air conditioning equipment and air conditioning furnaces who are in a position to offer the trade matched, factory engineered units of proved design and performance.

To such manufacturers, UNIVERSAL COOLER offers the most modern type of condensing units, in all sizes and capacities from ¾ h.p. to 15 h.p., painted any color and carrying your own name plate. Backed by 16 years' experience in building commercial electric refrigerating machinery for many of the leading manufacturers in the industry, we are exceptionally well fitted to assist in the engineering, selection and arrangement of refrigerating units to match your own equipment. Make our factory . . . your factory! For complete details, write to UNIVERSAL COOLER CORPORATION, Commercial Sales Dept., Detroit, Michigan. In Canada: Universal Cooler Company of Canada, Ltd., Brantford, Ontario.

# UNIVERSAL COOLER

***Suppliers of quality refrigerating machines to the air conditioning industry***




**The Patterson Indestructo COPPER-LINED Hot Water Heater in the Majestic Apartments overlooking Central Park in New York, can never rust and cause red water.**

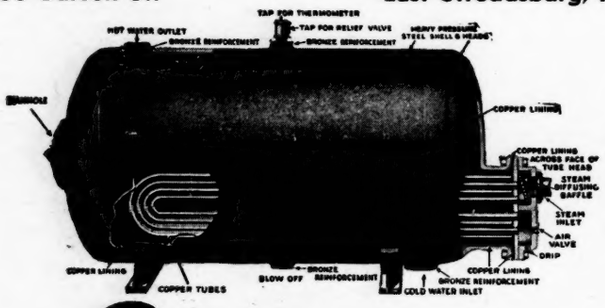
**Moreover, it will not deteriorate from rust, therefore will outwear many ordinary heaters and yield greater returns on the investment, when measured by actual cost over years of service. It is not always easy to see quality, but lack of quality cannot fool time.**

**Be safe and install Patterson Copper-Lined or Everdur Heaters. Then, if necessary, you can always fall back on our guarantee that these heaters will not rust or cause red water and that they will deliver all the hot water required, as hot as required and as fast as required.**

**Write for our interesting catalog.**

**The Patterson-Kelley Co., Inc.**

**105 Burson St. East Stroudsburg, Pa.**




**Patterson**

**Copper-Lined and Everdur Hot Water Heaters**

*(Concluded from page 71)*

misleading and unfair propaganda as this article represents.

There is an old saying that when it comes to selling public institutions, it does not matter where an incident took place, when or how, or if it ever took place, so long as the records show that it took place. The taxpayers are gullible.—*Len F. Sargent, Wausau Furnace and Parts Co., Wausau, Wis.*

* * *

In your February issue the article headed "School Heating Costs Cut 76%" is, in my judgment, a misapplication of the facts, and results in a misrepresentation as to the fuel economy of different types of equipment.

In the next to the last paragraph, there is this conclusion: "A saving of $830 in favor of the steam heating plant."

I maintain that that is not a correct statement, for the following reasons:

First, part of the saving is due to the fact that they are buying cheaper fuel for the steam heating plant—being a modern plant instead of an obsolete one, it of course could be equipped to burn the cheaper fuel. But that is not in favor of the *steam heating plant;* it is in favor of *modern equipment.*

Second, like every other split system, the radiation installed to heat the rooms makes the operation of the ventilating system unnecessary, heats the rooms by recirculation, and a big part of the actual saving in the amount of fuel required is due to the change from the old outmoded theory of 30 c.f.m. per person from out-of-doors as a basis of heating, to the modern theory of recirculation as the basis of heating—with ventilating air taken under control from out-of-doors.

On that basis, a furnace system, properly designed and of proper capacity, will be more economical in fuel than the steam system.

In spite of the tremendous saving shown in this comparison, the fuel bill is still not an economical one. The high school in Valley Park, Mo., having 14 rooms and a gymnasium, heated with a furnace system, with provision to take air from out-of-doors when and as needed, required only 34 tons of coal for the winter 1934-35 (soft coal), against the 62.6 tons of anthracite required for the Niagara Falls building for the year 1935-36.

The inference of the entire article is that steam is more economical than the furnace fan system. Such is not the case and there is nothing in this experience to prove it. I ask that you correct that inference.—*E. K. Campbell, E. K. Campbell Heating Co., Kansas City, Mo.*

## WITH THE MANUFACTURERS

*Anthracite Industries, Inc.,* Chrysler Building, New York, announces that *G. Gordon Cook* has been elected vice-president and treasurer of that organization and not vice-president and secretary as incorrectly reported on these pages recently.

*Carnegie-Illinois Steel Corp.,* Pittsburgh, Pa., has appointed *G. Reed Schreiner* advertising manager, succeeding C. R. Moffatt, recently appointed director of advertising of U. S. Steel Corp. of Delaware.

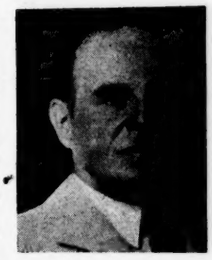
*Burnham Boiler Corp.*, Irvington, N. Y., has appointed *Warren Doble*, 220 Bush St., San Francisco, Calif., as its representative in northern California, including Monterey, Kings, Tulare and Inyo counties. *Southern California Engineering Co.*, 174 S. Western Ave., Los Angeles, will handle the southern part of the state, including San Luis Obispo, Kern and San Bernardino counties.

*Fairbanks, Morse & Co.*, Chicago, has moved its general offices from 900 S. Wabash Ave. to the modernized Fairbanks-Morse building at *600 S. Michigan.*

*Iron Fireman Manufacturing Co.*, Portland, Oreg., is entering the oil burner market. The company plans to introduce its new oil burners at its 1938 sales conventions, scheduled as follows: Cleveland, February 28 and March 1; New York, March 3-4; Atlanta, March 7-8; Chicago, March 10-11; St. Paul, March 14-15. The Iron Fireman oil burner is being offered nationally after being developed in the Pacific Northwest, where it has been sold under the name General Burner Association, Inc., Oil Burner. The oil burner will sell in the highest price class.

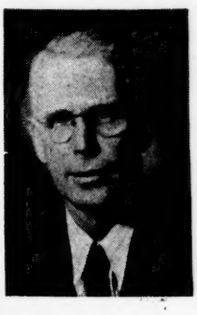
*Johnson Service Co.*, Milwaukee, Wis., has named *Joseph A. Cutler* president and general manager to succeed *H. W. Ellis*, who retired February 1. Associated with Johnson for 43 years, Mr. Ellis assumes the newly-created position of chairman of the board of directors. Formerly vice-president and general sales manager in charge of the Chicago office, Mr. Cutler will now be located in Milwaukee. He is succeeded in Chicago by *O. G. Ward*, formerly manager of the Denver office. *A. W. Cooper*, of the Los Angeles office, replaces Mr. Ward in the Denver office. *M. F. Rather*, formerly manager of the Cleveland office, has been appointed manager of the eastern district in New York following the retirement of Kennedy Duff. Mr. Rather is succeeded in Cleveland by *P. D. Gayman*, formerly Indianapolis manager. *Hugh V. Alexander*, sales engineer from Buffalo, becomes manager at Indianapolis.

*Joseph A. Cutler*

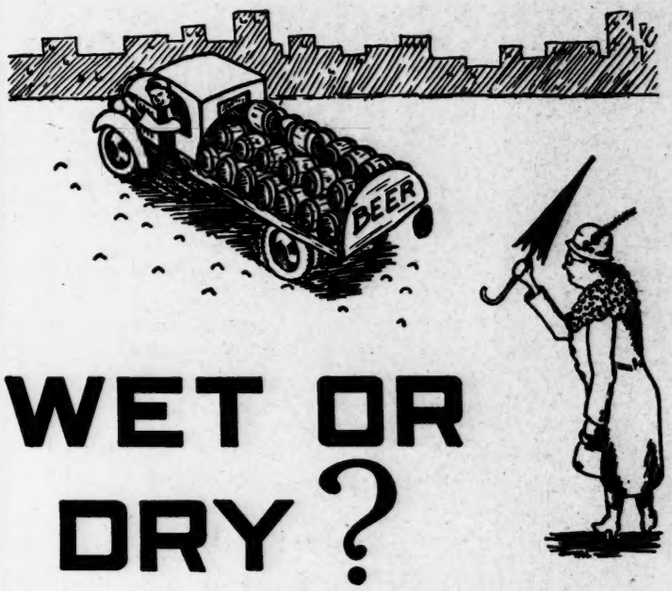


*H. W. Ellis*

*Modine Manufacturing Co.*, Racine, Wis., has appointed *The Paul R. Winston Company* of Dallas its representative in that territory. Mr. Winston, for 11 years prior to 1930, was connected with the engineering department of Briggs-Weaver Machinery Co. Since 1930 he has been active as a manufacturer's representative. *Disney-Leffel Co., Inc.*, Kansas City, is representing Modine in the western half of Missouri and the eastern half of Kansas. *Ace Combustion Engineering Co.* is Modine representative in Shreveport, La.



*Paul Winston*


WET OR DRY?

**No matter what the law may say on "suds" or high-balls, one place that must always be voted DRY is an underground steam line. There a tight, dry conduit installation is all-important. Only with such a job—adequately engineered, properly drained and insulated—will you receive maximum efficiency (over 90%) from your steam lines. In the opinion of most qualified engineers, that means a Ric-wiL Tile or Cast Iron System, insulated with genuine waterproof asbestos Dry-paC. Get the FACTS—then act accordingly.**

Ric-wiL comes in various designs in standard or extra-heavy Tile, and also in Cast Iron. Ric-wiL keeps steam lines over 90% efficient. Basic features are closed construction, adequate under-drainage, Loc-liP side joints, improved pipe supports, and certified Dry-paC waterproof asbestos insulation. Sectional pipe covering may be used if desired. Ric-wiL service covers engineering, manufacture, and installation and is nation-wide. Write for latest Bulletin and Dry-paC sample.

**The RIC-WIL Co., Union Trust Bldg., Cleveland, Ohio**

New York San Francisco Chicago

*Agents in principal cities*

Standard Ric-wiL conduit is assembled on the job—materials and workmanship completely open to inspection.

REGISTERED IN U. S. PATENT OFFICE

RIC-WIL

CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

**Stop VIBRATION NOISE**

**in Compressor Lines**

**Rex Vibra-Sorbers absorb vibration, stop noise, insure permanently quiet operation. Sweated into the compressor lines, they act as flexible couplings for speedy, trouble-free installation. All sizes available.**

**VIBRA-SORBERS**

Made by
CHICAGO METAL HOSE CORPORATION
Maywood, Illinois

**STOPS OVERHEATED WATER**

***Reduces Fuel Losses***

**POWERS Hot Water Tank Regulator**

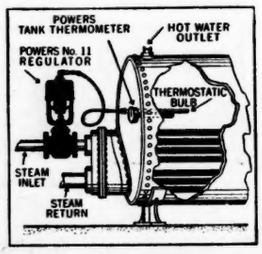
Ends Hot Water Complaints

***Pays back its cost several times a year***

OVERHEATED water causes complaints, wastes fuel — shortens life of valves and plumbing fixtures and increases deposit of lime in pipes. • Powers regulators will help to reduce these losses. Install them on your hot water heaters. They keep the water at the right temperature. Fuel savings alone often pay back their cost several times a year. As they usually last 10 to 15 years they pay big dividends. *Write for Bulletin 2035.*

**The Powers Regulator Co.** Offices in 45 Cities. See your phone directory. 2718 Greenview Avenue, Chicago; 231 E. 46th Street, New York City; 1808 West 8th Street, Los Angeles.

*45 Years of Temperature Control*




**POWERS**
AUTOMATIC
WATER TEMPERATURE CONTROL


*South Bend Air Products, Inc.*, South Bend, Ind. was recently organized to purchase and continue the ventilating equipment business formerly conducted by the industrial fan division of the Bendix Products Corp. *Marshall A. Smith, Jr.*, is president and general manager of the new company. Mr. Smith was formerly in charge of sales and engineering of the Bendix industrial fan division. The complete line of fans and other equipment is being continued.

*Howard R. Stevens*, formerly associated with Detroit Stoker Company in Boston, has opened an office at 10 High St., Boston. He is handling all types of fuel burning equipment.

## NEW CATALOGS

*(Unless otherwise indicated the publishers of catalogs will supply copies on request without charge.)*

*Air Meters.* A double-page bulletin describing the Detroit line of grillometers, draft gages, and pitot tubes. The grillometer is a direct reading vane type air meter for measuring air velocities from 50 to 3500 f.p.m. Detroit Air Meter Co., P. O. Box 1473, Detroit.

*Copper Tube.* A standard-size, 28-page publication, C-2, on copper tubes and fittings for heating lines. Gives information on gages, types and tempers of tubes, wrought and cast fittings, methods of assembly, means of providing for expansion and contraction and comparative heat losses. One chapter suggests the procedure for sizing copper tubes. The American Brass Co., Waterbury, Conn.

*Electric Motors.* A standard-size, 12-page bulletin entitled "Electric Drive Selector and Hints on Maintenance." Pamphlet describes and illustrates the requirements which must be met in selecting a motor. A chart shows how to select a motor, gives types of motors, applications and descriptions, and gives concrete examples of installations. Another chart shows how to select the control and circuit protection. Whether the control is to be manual or magnetic, the motor for each control is indicated. A section of the pamphlet is devoted to maintenance, and there are discussed 10 hints on preventive maintenance. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

*Filters.* A 32-page catalog on Protectomotor air filters. Catalog is divided into two sections, the first dealing with industrial filters for air compressors, pumps and internal combustion engines, while the second part deals with ventilating filters. Included in the second part are descriptions and data on the Protectomotor panel filter—a cleanable dry type filter; the Multi-V-Type air filter—another cleanable dry type filter; the Protectovent window ventilating unit—a portable air moving, filtering and silencing device; and the Staynew automatic oil impingement, self-cleaning air filter. Staynew Filter Corp., Rochester, N. Y.

*Filters.* A standard-size, 14-page bulletin, No. 500-3, illustrating and describing the "Norblo" square type air filters. These filters are automatic and designed for continuous service. Filters have a shaking mechanism which is used for cleaning the bags. The operation of the mechanism may be controlled by an electrical

timing device. Filtering medium may be of cotton, wool, glass or asbestos bags, depending on the nature of the job. THE NORTHERN BLOWER CO., WEST 65TH ST., CLEVELAND, OHIO.

*Humidity Instruments.* A standard-size, 28-page catalog, No. 6502, describing the Brown instrument line of hygrometers for recording and controlling humidity. Describes the features and operating principles of the Brown recording hygrometer and psychrometers and gives illustrations and specifications of the various types of instruments. THE BROWN INSTRUMENT CO., PHILADELPHIA, PA.

*Steam Traps.* A standard size, 12-page bulletin, T-1733, describing the Yarway impulse steam trap. Bulletin gives complete information on the construction of the trap, how the trap works, how to select impulse steam traps, and gives various typical piping diagrams. YARNELL-WARING CO., CHESTNUT HILL, PHILADELPHIA, PA.

*Thermometers.* A four-page folder on the new Brown resistance thermometer. This folder explains how the Brown resistance thermometer furnishes a positive means of checking the correct temperature throughout an entire building from one central point. BROWN INSTRUMENT CO., SUBSIDIARY OF MINNEAPOLIS-HONEYWELL REGULATOR CO., MINNEAPOLIS, MINN.

## COMING EVENTS

**MARCH 18-27, 1938. National Homes Exposition, Madison Square Garden, New York. Madison Square Garden Corp., 8th Ave. at 49th and 50th Sts., New York.**

**MARCH 28-31, 1938. 8th Annual Forced Warm Air Short Course, Michigan State College, East Lansing Mich. Prof. L. G. Miller, Mechanical Engineering Department. Michigan State College, East Lansing, Mich.**

**APRIL 11-16, 1938. Mechanical Equipment Show, Textile Hall, Greenville, S. Car. William G. Sirrine, President. Textile Hall Corp., Greenville, S. Car.**

**APRIL 13-15, 1938. Midwest Power Conference, La Salle Hotel, Chicago. Sponsored by Armour Institute of Technology. Director, L. E. Gointer, Dean of Graduate Division.**

**MAY 31-JUNE 2, 1938. 49th Annual Convention, Heating, Piping and Air Conditioning Contractors National Association. Boston, Mass.**

**JUNE 20-22, 1938. Semi-Annual Meeting, American Society of Heating and Ventilating Engineers, Hot Springs, Va.**

**JUNE 20-23, 1938. 31st Annual Convention, National Association of Building Owners and Managers. Hotel Schroeder, Milwaukee, Wis. Association headquarters, 134 S. LaSalle St., Chicago.**

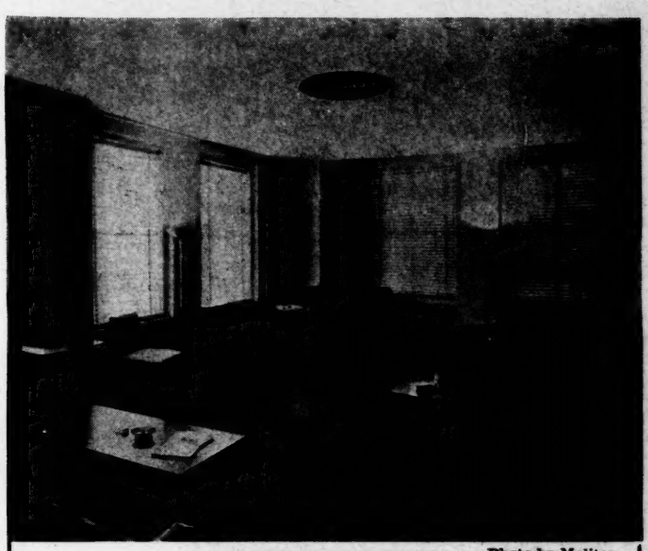
**JUNE 20-24, 1938. 25th Spring Meeting, American Society of Refrigerating Engineers, The Pennsylvania State College, State College, Pa.**

**JUNE 28-JULY 1, 1938. Annual Convention and Manufacturers' Exhibit, National District Heating Association, Lord Baltimore Hotel, Baltimore, Md. Wm. H Sanford, Secretary-Treasurer, NDHA, 1317 Spruce St., Philadelphia, Pa.**

**SEPTEMBER 21-23, 1938. 16th Annual Conference, National Industrial Advertisers Association, Cleveland, Ohio. Stanley Knisely, Advertising Manager, Republic Steel Corp., Cleveland, Ohio.**

**DECEMBER 5-10, 1938. 13th National Exposition of Power and Mechanical Engineering, Grand Central Palace, New York.**

**JANUARY 23-27, 1939. Annual Meeting, American Society of Heating and Ventilating Engineers, Pittsburgh, Pa**

AN EXECUTIVE OFFICE Photo by Molitor

*Engineers and Contractors*—ALLEN ENGINEERING CORP.

**Draftless Year-'round AIR CONDITIONING at Columbia Broadcasting System, Inc.**

The success of draftless AIR DISTRIBUTION by means of ANEMOSTATS in the studios enabling the artists to perform in perfect cool comfort, without unpleasant and harmful drafts, resulted in the use of ANEMOSTATS in the air conditioning installation for the company's executive offices on the 20th floor.

The ANEMOSTAT is an accepted scientific achievement. It has solved major problems of the Air Conditioning Industry by reducing the speed of air flow to such an extent that it is no longer perceptible as a draft. The ANEMOSTAT mixes room air with conditioned air within the device. Thus the predetermined room temperature is established considerably above the breathing level. The ANEMOSTAT permits greater temperature differentials, smaller ducts, smaller plants, lower installation and operating costs. The ANEMOSTAT can be applied to existing systems as well as to new installations.

**INSTALLATION DATA TYPE "C" COMPLETE AIR CONDITIONING SYSTEM**
(SUMMER AND WINTER)

| | |
|---|---|
| Length of Room ..... 23 | Width ..... 15' |
| Ceiling Height ..... 8' 6" | Neck diameter ..... 10" |
| Velocity in Anemostat neck ..... | 900 f.p.m |
| Volume of air distributed through Anemostat ..... | 500 c.f.m. |
| Temperature differential ..... | 10° to 15° |
| Largest cone diameter ..... | 30" |

**ANEMOSTAT CORP. OF AMERICA**
**10 East 39th Street, New York City**

ANEMOSTAT
"No Air Conditioning System is better than its Air Distribution"

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